Handbook of Chemical Processing Equipment

Nicholas P. Cheremisinoff, Ph.D.

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PREFACE

The chemical industry represents a 455-billion-dollar-a-year business, with products ranging from cosmetics, to fuel products, to plastics, to pharmaceuticals, health care products, food additives, and many others. It is diverse and dynamic, with market sectors rapidly expanding, and in turmoil in many parts of the world. Across these varied industry sectors, basic unit operations and equipment are applied on a daily basis, and indeed although there have been major technological innovations to processes, many pieces of equipment are based upon a foundation of engineering principles developed more than 50 years ago.

The *Handbook of Chemical Processing Equipment* has been written as a basic reference for process engineers. It provides practical information on the working principles and engineering basis for major equipment commonly used throughout the chemical processing and allied industries. Although written largely with the chemical engineer in mind, the book's contents are general enough, with sufficient background and principles described, that other manufacturing and process engineers will find it useful.

The handbook is organized into eight chapters. Chapters 1 through 3 deal with heat transfer equipment used in a variety of industry applications ranging from process heat exchange, to evaporative cooling, to drying and solvent recovery applications, humidity control, crystallization, and others. Chapters 4 and 5 cover stagewise mass transfer equipment. Specifically, Chapter 4 covers distillation, and Chapter 5 covers classical mass transfer equipment involving absorption, adsorption, extraction, and membrane technologies. Chapter 6 discusses equipment used in mass separation based upon physical or mechanical means. This includes such equipment topics as sedimentation, centrifugal separation, filtrations methods. Chapter 7 covers mixing equipment and various continuous contacting devices such as gas-solids fluidized beds. Finally, Chapter 8 provides the reader with a compendium of short calculation methods for commonly encountered process operations. The calculation methods are readily set up on a personal computer's standard software spreadsheet.

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Select references are provided in each chapter for more in-depth coverage of an equipment subject, including key Web sites that offer vendor-specific information and equipment selection criteria. In a number of chapters, sample calculations are provided to guide the reader through the use of design and scale-up formulas that are useful in preparing equipment specifications or in establishing preliminary designs.

Although the author has taken great care to ensure that the information presented in this volume is accurate, neither he nor the publisher will endorse or guarantee any designs based upon materials provided herein. The author wishes to thank Butterworth-Heinemann Publishers for their fine production of this volume.

Nicholas P. Cheremisinoff, Ph.D.

Chapter 1
HEAT EXCHANGE EQUIPMENT

INTRODUCTION

Prior to the 19th century, it was believed that the sense of how hot or cold an object felt was determined by how much "heat" it contained. Heat was envisioned as a liquid that flowed from a hotter to a colder object; this weightless fluid was called "caloric", and until the writings of Joseph Black (1728-1799), no distinction was made between heat and temperature. Black distinguished between the quantity (caloric) and the intensity (temperature) of heat. Benjamin Thomson, Count Rumford, published a paper in 1798 entitled "An Inquiry Concerning the Source of Heat which is Excited by Friction". Rumford had noticed the large amount of heat generated when a cannon was drilled. He doubted that a material substance was flowing into the cannon and concluded "it appears to me to be extremely difficult if not impossible to form any distinct idea of anything capable of being excited and communicated in the manner the heat was excited and communicated in these experiments except motion."

But it was not until J. P. Joule published a definitive paper in 1847 that the caloric idea was abandoned. Joule conclusively showed that heat was a form of energy. As a result of the experiments of Rumford, Joule, and others, it was demonstrated (explicitly stated by Helmholtz in 1847), that the various forms of energy can be transformed one into another.

When heat is transformed into any other form of energy, or when other forms of energy are transformed into heat, the total amount of energy (heat plus other forms) in the system is constant. This is known as the first law of thermodynamics, i.e., the conservation of energy. To express it another way: it is in no way possible either by mechanical, thermal, chemical, or other means, to obtain a perpetual motion machine; i.e., one that creates its own energy.

A second statement may also be made about how machines operate. A steam engine uses a source of heat to produce work. Is it possible to completely convert the heat energy into work, making it a 100% efficient machine? The answer is to be found in the second law of thermodynamics: No cyclic machine can convert heat energy wholly into other forms of energy. It is not possible to construct a cyclic machine that does nothing, but withdraw heat energy and convert it into mechanical energy. The second law of thermodynamics implies the irreversibility

of certain processes - that of converting all heat into mechanical energy, although it is possible to have a cyclic machine that does nothing but convert mechanical energy into heat.

Sadi Carnot (1796-1832) conducted theoretical studies of the efficiencies of heat engines (a machine which converts some of its heat into useful work). He was trying to model the most efficient heat engine possible. His theoretical work provided the basis for practical improvements in the steam engine and also laid the foundations of thermodynamics. He described an ideal engine, called the Carnot engine, that is the most efficient way an engine can be constructed. He showed that the efficiency of such an engine is given by:

$$\text{efficiency} = 1 - T''/T'$$

where the temperatures, T' and T", are the cold and hot "reservoirs", respectively, between which the machine operates. On this temperature scale, a heat engine whose coldest reservoir is zero degrees would operate with 100% efficiency. This is one definition of absolute zero. The temperature scale is called the absolute, the thermodynamic , or the kelvin scale.

The way, that the gas temperature scale and the thermodynamic temperature scale are shown to be identical, is based on the microscopic interpretation of temperature, which postulates that the macroscopic measurable quantity called temperature, is a result of the random motions of the microscopic particles that make up a system.

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About the same time that thermodynamics was evolving, James Clerk Maxwell (1831-1879) and Ludwig Boltzmann (1844-1906) developed a theory, describing the way molecules moved - molecular dynamics. The molecules that make up a perfect gas move about, colliding with each other like billiard balls and bouncing off the surface of the container holding the gas. The energy, associated with motion, is called Kinetic Energy and this kinetic approach to the behavior of ideal gases led to an interpretation of the concept of temperature on a microscopic scale.

The amount of kinetic energy each molecule has is a function of its velocity; for the large number of molecules in a gas (even at low pressure), there should be a range of velocities at any instant of time. The magnitude of the velocities of the various particles should vary greatly; no two particles should be expected to have the exact same velocity. Some may be moving very fast; others - quite slowly.

Maxwell found that he could represent the distribution of velocities statistically by a function, known as the Maxwellian distribution. The collisions of the molecules with their container gives rise to the pressure of the gas. By considering the average force exerted by the molecular collisions on the wall, Boltzmann was able to show that the average kinetic energy of the molecules was

directly comparable to the measured pressure, and the greater the average kinetic energy, the greater the pressure.

From Boyles' Law, it is known that the pressure is directly proportional to the temperature, therefore, it was shown that the kinetic energy of the molecules related directly to the temperature of the gas. A simple thermodynamic relation holds for this:

$$\text{average kinetic energy of molecules} = 3kT/2$$

where k is the Boltzmann constant. Temperature is a measure of the energy of thermal motion and, at a temperature of zero, the energy reaches a minimum (quantum mechanically, the zero-point motion remains at 0 °K).

About 1902, J. W. Gibbs (1839-1903) introduced statistical mechanics with which he demonstrated how average values of the properties of a system could be predicted from an analysis of the most probable values of these properties found from a large number of identical systems (called an ensemble). Again, in the statistical mechanical interpretation of thermodynamics, the key parameter is identified with a temperature, which can be directly linked to the thermodynamic temperature, with the temperature of Maxwell's distribution, and with the perfect gas law.

Temperature becomes a quantity definable either in terms of macroscopic thermodynamic quantities, such as heat and work, or, with equal validity and identical results, in terms of a quantity, which characterized the energy distribution among the particles in a system. With this understanding of the concept of temperature, it is possible to explain how heat (thermal energy) flows from one body to another.

Thermal energy is carried by the molecules in the form of their motions and some of it, through molecular collisions, is transferred to molecules of a second object, when put in contact with it. This mechanism for transferring thermal energy is called conduction.

A second mechanism of heat transport is illustrated by a pot of water set to boil on a stove - hotter water closest to the flame will rise to mix with cooler water near the top of the pot. Convection involves the bodily movement of the more energetic molecules in a liquid or gas. The third way, that heat energy can be transferred from one body to another, is by radiation; this is the way that the sun warms the earth. The radiation flows from the sun to the earth, where some of it is absorbed, heating the surface.

These historical and fundamental concepts form the foundation for the design, applications, and operations of a major class of equipment that are used throughout the chemical process industries - heat exchange equipment, or heat exchangers. There are many variations of these equipment and a multitude of

applications. However, the design configurations for these equipment are universal, meaning that they generally are not specific to a particular industry sector. In the United States in 1998, the chemical process industries (CPI) invested more than $700 million in capital equipment related to heat transfer. Much of that investment was driven by a growing body of environmental legislation, such as the U.S. Clean Air Act Amendments. The use of vent condensers, for example, which use heat exchangers to reduce the volume of stack emissions, is increasing. Heat exchanger makers have responded to growing environmental concerns over fugitive emissions, as well by developing a new breed of leak-tight heat exchangers, designed to keep process fluids from leaking and volatile organic compounds from escaping to the atmosphere. Gasketed exchangers are benefitting from improvements in the quality and diversity of elastomer materials and gasket designs. The use of exchangers with welded connections, rather than gaskets, is also reducing the likelihood of process fluid escape. Throughout the 1990's, the use of heat exchangers has expanded into non-traditional applications. This, coupled with a variety of design innovations, has given chemical engineers a wider variety of heat exchanger options to choose from than ever before. Operating conditions, ease of access for inspection and maintenance, and compatibility with process fluids are just some of the variables CPI engineers must consider when assessing heat exchanger options. Other factors include: maximum design pressure and temperature, heating or cooling applications, maintenance requirements, material compatibility with process fluids, gasket compatibility with process fluids, cleanliness of the streams, and temperature approach. This chapter provides an overview of the most commonly employed equipment. Emphasis is given to practical features of these systems, and typical examples of industrial applications are discussed.

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GENERAL CONCEPTS OF HEAT TRANSFER

Before discussing typical equipment commonly used throughout the chemical processing industries, some general concepts and definitions regarding the subject of heat transfer are reviewed. The term **heat** in physics, refers to the transfer of energy from one part of a substance to another, or from one object to another, because of a difference in temperature. Heat flows from a substance at a higher temperature to a substance at a lower temperature, provided the volume of the objects remains constant. Heat does not flow from a lower to a higher temperature, unless another form of energy transfer, work, is also present.

Until the beginning of the 19th century, it was thought that heat was an invisible substance called caloric. An object at a high temperature was thought to contain more caloric than one at a low temperature. However, British physicist Benjamin Thompson in 1798 and British chemist Sir Humphry Davy in 1799 presented

evidence that heat, like work, is a form of energy transfer. In a series of experiments between 1840 and 1849, British physicist James Prescott Joule provided conclusive evidence that heat is a form of energy in transit, and that it can cause the same changes as work.

The sensation of warmth or coldness is caused by temperature. Adding heat to a substance not only raises its temperature, but also produces changes in several other qualities. The substance expands or contracts; its electric resistance changes; and in the gaseous form, its pressure changes. Five different temperature scales are in use today: Celsius, Fahrenheit, Kelvin, Rankine, and international thermodynamic.

The term **resistance** refers to the property of any object or substance to resist or oppose the flow of an electrical current. The unit of resistance is the ohm. The abbreviation for electric resistance is R and the symbol for ohms is the Greek letter omega, Ω. For certain electrical calculations the reciprocal of resistance is used, 1/R, which is termed conductance, G. The unit of conductance is the mho, or ohm spelled backward, and the symbol is an inverted omega.

Pressure, in mechanics, is the force per unit area exerted by a liquid or gas on an object or surface, with the force acting at right angles to the surface and equally in all directions. In the United States, pressure is usually measured in pounds per square inch (psi); in international usage, in kilograms per square centimeters, or in atmospheres; and in the international metric system (SI), in newtons per square meter (International System of Units). Most pressure gauges record the difference between a fluid pressure and local atmospheric pressure. Types of common pressure gauges include U-tube manometers, for measuring small pressure differences; Bourdon gauges, for measuring higher pressure differences; gauges that use piezoelectric or electrostatic sensing elements, for recording rapidly changing pressures; McLeod gauges, for measuring very low gas pressures; and gauges that use radiation, ionization, or molecular effects to measure low gas pressures (in vacuum technology). In the atmosphere the decreasing weight of the air column with altitude leads to a reduction in local atmospheric pressure. Partial pressure is the effective pressure that a single gas exerts in a mixture of gases. In the atmosphere the total pressure is equal to the sum of the partial pressures.

Heat is measured in terms of the calorie, defined as the amount of heat necessary to raise the temperature of 1 gram of water at a pressure of 1 atmosphere from 15° to 16 °C. This unit is sometimes called the small calorie, or gram calorie, to distinguish it from the large calorie, or kilocalorie, equal to 1000 small calories, which is used in nutritional studies. In mechanical engineering practice in the United States and the United Kingdom, heat is measured in British thermal units (Btu). One Btu is the quantity of heat required to raise the temperature of 1 pound of water 1° F and is equal to 252 calories.

The term **latent heat** is also pertinent to our discussions. The process of changing from solid to gas is referred to as sublimation; from solid to liquid, as melting; and from liquid to vapor, as vaporization. The amount of heat required to produce such a change of phase is called latent heat. If water is boiled in an open container at a pressure of 1 atmosphere, its temperature does not rise above 100° C (212° F), no matter how much heat is added. The heat that is absorbed without changing the temperature is latent heat; it is not lost, but is expended in changing the water to steam.

The **phase rule** is a mathematical expression that describes the behavior of chemical systems in equilibrium. A chemical system is any combination of chemical substances. The substances exist as gas, liquid, or solid phases. The phase rule applies only to systems, called heterogeneous systems, in which two or more distinct phases are in equilibrium. A system cannot contain more than one gas phase, but can contain any number of liquid and solid phases. An alloy of copper and nickel, for example, contains two solid phases. The rule makes possible the simple correlation of very large quantities of physical data and limited prediction of the behavior of chemical systems. It is used particularly in alloy preparation, in chemical engineering, and in geology.

The subject of heat transfer refers to the process by which energy in the form of heat is exchanged between objects, or parts of the same object, at different temperatures. Heat is generally transferred by radiation, convection, or conduction, processes that may occur simultaneously.

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Conduction is the only method of heat transfer in opaque solids. If the temperature at one end of a metal rod is raised, heat travels to the colder end. The mechanism of conduction in solids is believed to be partially due to the motion of free electrons in the solid matter. This theory helps explain why good conductors of electricity also tend to be good conductors of heat. In 1882 French mathematician Jean Baptiste Joseph Fourier formulated a law that the rate, at which heat is conducted through an area of an object, is proportional to the negative of the temperature change through the object. Conduction also occurs between two objects, if they are brought into contact. Conduction between a solid surface and a moving liquid or gas is called convection. The motion of the fluid may be natural or forced. If a liquid or gas is heated, its mass per unit of volume generally decreases. If the substance is in a gravitational field, the hotter, lighter fluid rises while the colder, heavier fluid sinks. This kind of motion is called natural **convection**. Forced convection is achieved by putting the fluid between different pressures, and so forcing motion to occur according to the law of fluid mechanics.

Radiation is a process that is different from both conduction and convection, because the substances exchanging heat need not be touching and can even be separated by a vacuum. A law formulated by German physicist Max Planck in

1900 states, in part, that all substances emit radiant energy, simply because they have a positive absolute temperature. The higher the temperature, the greater the amount of energy emitted. In addition to emitting, all substances are capable of absorbing radiation. The absorbing, reflecting, and transmitting qualities of a substance depend upon the wavelength of the radiation.

In addition to heat transfer processes that result in raising or lowering temperatures, heat transfer can also produce phase changes in a substance, such as the melting of ice. In engineering, heat transfer processes are usually designed to take advantage of this ability. For instance, a space capsule reentering the atmosphere at very high speeds is provided with a heat shield that melts to prevent overheating of the capsule's interior. The frictional heat, produced by the atmosphere, is used to turn the shield from solid to liquid and does not raise the temperature of the capsule.

Evaporation is the gradual change of a liquid into a gas without boiling. The molecules of any liquid are constantly moving. The average molecular speed depends on the temperature, but individual molecules may be moving much faster or slower than the average. At temperatures below the boiling point, faster molecules approaching the liquid's surface may have enough energy to escape as gas molecules. Because only the faster molecules escape, the average speed of the remaining molecules decreases, lowering the liquid's temperature, which depends on the average speed of the molecules.

An additional topic to discuss from an introductory standpoint is **thermal insulating materials**. These materials are used to reduce the flow of heat between hot and cold regions. The sheathing often placed around steam and hot-water pipes, for instance, reduces heat loss to the surroundings, and insulation placed in the walls of a refrigerator reduces heat flow into the unit and permits it to stay cold.

Thermal insulation generally has to fulfill one or more of three functions: to reduce thermal conduction in the material where heat is transferred by molecular or electronic action; to reduce thermal convection currents, which can be set up in air or liquid spaces; and to reduce radiation heat transfer where thermal energy is transported by electromagnetic waves. Conduction and convection can be suppressed in a vacuum, where radiation becomes the only method of transferring heat. If the surfaces are made highly reflective, radiation can also be reduced. As examples, thin aluminum foil can be used in building walls, and reflecting metal on roofs minimizes the heating effect of the sun. Thermos bottles or Dewar flasks provide insulation through an evacuated double-wall arrangement in which the walls have reflective silver or aluminum coatings. Air offers resistance to heat flow at a rate about 15,000 times higher than that of a good thermal conductor, such as silver, and about 30 times higher than that of glass.

Typical insulating materials, therefore, are usually made of nonmetallic materials and are filled with small air pockets. They include magnesium carbonate, cork, felt, cotton batting, rock or glass wool, and diatomaceous earth. Asbestos was once widely used for insulation, but it has been found to be a health hazard and has, therefore, been banned in new construction in the U.S.

In building materials, air pockets provide additional insulation in hollow glass bricks, insulating or thermopane glass (two or three sealed glass panes with a thin air space between them), and partially hollow concrete tile. Insulating properties are reduced, if the air space becomes large enough to allow thermal convection, or, if moisture seeps in and acts as a conductor. The insulating property of dry clothing, for example, is the result of air entrapped between the fibers; this ability to insulate can be significantly reduced by moisture. Home-heating and air-conditioning costs can be reduced by proper building insulation. In cold climates about 8 cm (about 3 in.) of wall insulation and about 15 to 23 cm (about 6 to 9 in.) of ceiling insulation are recommended. The effective resistance to heat flow is conventionally expressed by its R-value (resistance value), which should be about 11 for wall and 19 to 31 for ceiling insulation.

Superinsulation has been developed, primarily for use in space, where protection is needed against external temperatures near absolute zero. Superinsulation fabric consists of multiple sheets of aluminized mylar, each about 0.005 cm (about 0.002 in.) thick, and separated by thin spacers with about 20 to 40 layers per cm (about 50 to 100 layers per in.).

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Governing Expressions for Heat Exchangers

When a hot fluid stream and a cold fluid stream, separated by a conducting wall, exchange heat, the heat that is transferred across a differential element can be represented by the following expression (refer to Figure 1):

$$dq = U\,\Delta t\,dA$$

where dq = heat transferred across differential element dA (W),

U = Overall heat transfer coefficient (W/m-°K),

Δt = temperature difference across element dA (°K),

dA = heat transfer area for the differential element (m^2).

The expression can be integrated over the entire heat exchanger using the simplification that the changes in U with temperature and position are negligible.

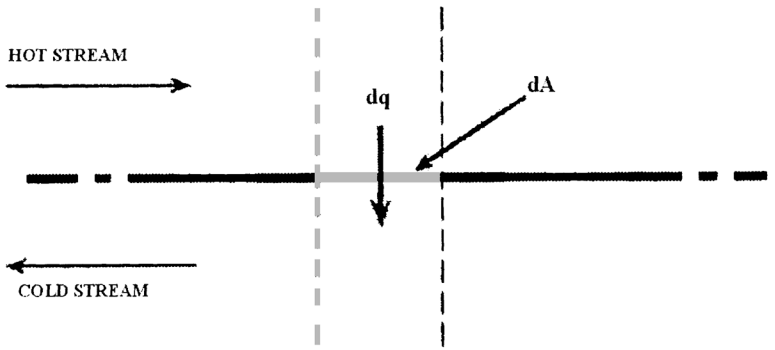

Figure 1. *Heat exchange across a differential element in a heat exchanger.*

In this manner, an average value of U can be applied to the whole exchanger. Ideally, the heat lost by the hot fluid stream is transferred totally to the cold stream, and hence, integrating results in the following expression:

$$q = UA\Delta t_{lm}$$

where A = the total heat exchange area (m^2), q = the total heat transferred (W), and U = the overall heat transfer coefficient, assumed to be constant throughout the exchanger (W/m^2-°K). The parameter Δt_{lm} is the log-mean temperature difference (in units of °K) and defined by the following expression:

$$\Delta t_{lm} = \Theta/\Gamma$$

where $\Theta = (t_{h,in} - t_{c,out}) - (t_{h,out} - t_{c,in})$

$\Gamma = \ln ((t_{h,in} - t_{c,out}t)/(t_{h,out} - t_{c,in}))$

The overall heat transfer coefficient, U, is a measure of the conductivity of all the materials between the hot and cold streams. For steady state heat transfer through the convective film on the outside of the exchanger pipe, across the pipe wall and through the convective film on the inside of the convective pipe, the overall heat transfer coefficient may be stated as:

$$1/U = A/h_1A_1 + A\Delta x/kA_{lm} + A/h_2A_2$$

where A = a reference area (m^2),

h_1 = heat transfer coefficient inside the pipe (W/m^2-°K),

A_1 = area inside the pipe (m^2),

Δx = pipe wall thickness (m),

k = thermal conductivity of the pipe (W/m-°K),

h_2 = heat transfer coefficient outside the pipe (W/m^2-°K),

A_2 = area outside the pipe (m^2).

The term A_{lm} is the log-mean area of the pipe (in m^2) defined as follows:

$$A_{lm} = (A_1 - A_2)/\ln(A_1/A_2)$$

Estimation of the heat transfer coefficients for forced convection of a fluid in pipes is usually based on empirical expressions. The most well known expression for this purpose is:

$$Nu = 0.023\ Re^{0.8}\ Pr^{0.33}$$

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where Nu is the Nusselt number, a dimensionless group defining the relative significance of the film heat transfer coefficient to the conductivity of the pipe wall, Re is the Reynolds number, which relates inertial forces to viscous forces and thereby characterizes the type of flow regime, and Pr is the Prandtl number, which relates the thermal properties of the fluid to the conductivity of the pipe.

It is well known from heat transfer studies that the fluid heat transfer coefficient, h_1, is proportional to the velocity, v, of the fluid raised to the power 0.8. If all other parameters are kept constant, it then follows that a plot of $1/v^{0.8}$ versus $1/U$ results in a straight line with an intercept, representing the sum of the vapor film conductance and the wall conductance. Knowing the wall conductance, the vapor film conductance can be determined from the intercept value. Many of the properties used in the empirical expression are functions of temperature. In general, the properties needed to evaluate the above empirical expression are taken at the mean bulk temperature of the fluid, i.e., the average between the inlet and outlet temperatures. For water however, a temperature correction must be applied. The temperature corrected plot for water would be $1/(1+0.011t)v^{0.8}$ versus $1/U$, where t is the average fluid temperature measured in °F. The resulting plot should be linear for each separate steam pressure, thereby producing a series of lines with the same slope, but having a different intercept, that is a function of pressure.

Another area to consider is heat exchanger efficiency. The concept of efficiency is to compare the actual performance of a piece of equipment with the ideal performance (i.e., the maximum potential heat transfer). The maximum heat transfer possible is established by the stream that has the minimum heat capacity. That is the minimum value for the product of stream mass flowrate and specific heat. This stream would, for maximum heat transfer, leave the exchanger at the inlet temperature of the other stream. In terms of the hot stream, the efficiency can be stated as:

$$e = C_{p,h}m_h(t_{h,in} - t_{h,out})/((C_pm)_{min}(t_{h,in} - t_{c,in}))$$

And, in terms of the cold stream:

$$e = C_{p,c}m_c(t_{c,in} - t_{c,out})/((C_pm)_{min}\,(t_{h,in} - t_{c,in}))$$

In the above expressions:

e = heat exchanger efficiency,

$t_{h,in}$ = the inlet temperature of the hot stream (°K),

$t_{c,out}$ = the outlet temperature of the cold stream (°K),

$t_{h,out}$ = the outlet temperature of the hot stream (°K),

$t_{c,in}$ = the inlet temperature of the cold stream (°K),

$C_{p,h}m$ = the product of the hot stream heat capacity and the mass flowrate,

$C_{p,c}m$ = the product of the cold stream heat capacity and the mass flowrate,

$(C_pm)_{min}$ = the minimum product of stream heat capacity and mass flowrate.

Knowing the efficiency, one can use this value to predict heat exchanger performance for other streams and fluids. Efficiency is based on the maximum amount of heat that can be transferred:

$$q = e(C_pm)_{min}\,(t_{h,in} - t_{c,in})$$

AIR COOLED HEAT EXCHANGERS

Air cooled heat exchangers are used to transfer heat from a process fluid to ambient air. The process fluid is contained within heat conducting tubes. Atmospheric air, which serves as the coolant, is caused to flow perpendicularly across the tubes in order to remove heat. In a typical air cooled heat exchanger, the ambient air is either forced or induced by a fan or fans to flow vertically across a horizontal section of tubes. For condensing applications, the bundle may be sloped or vertical. Similarly, for relatively small air cooled heat exchangers, the air flow may be horizontal across vertical tube bundles.

In order to improve the heat transfer characteristics of air cooled exchangers, the tubes are provided with external fins. These fins can result in a substantial increase in heat transfer surface. Parameters such as bundle length, width and number of tube rows vary with the particular application as well as the particular finned tube design.

The choice of whether air cooled exchangers should be used is essentially a question of economics including first costs or capital costs, operating and maintenance expenses, space requirements, and environmental considerations; and involves a decision weighing the advantages and disadvantages of cooling with air.

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The advantages of cooling with air may be seen by comparing air cooling with the alternative of cooling with water. The primary advantages and disadvantages of air cooled heat exchangers are summarized in Table 1. These issues should be examined on a case by case basis to assess whether air cooled systems are economical and practical for the intended application. Specific systems are described later in this chapter. The major components of air cooled heat exchangers include the finned tube, the tube bundle, the fan and drive assembly, an air plenum chamber, and the overall structural assembly. Each component is briefly described below.

Finned Tubes

Common to all air cooled heat exchangers is the tube, through which the process fluid flows. To compensate for the poor heat transfer properties of air, which flows across the outside of the tube, and to reduce the overall dimensions of the heat exchanger, external fins are added to the outside of the tube. A wide variety of finned tube types are available for use in air cooled exchangers. These vary in geometry, materials, and methods of construction, which affect both air side thermal performance and air side pressure drop. In addition, particular

combinations of materials and/or fin bonding methods may determine maximum design temperature limitations for the tube and limit environments, in which the tube might be used. The use of a particular fin tube is essentially a matter of agreement between the air cooled heat exchanger manufacturer and the user. Finned tubes may differ in the means, by which the fins themselves are attached or bonded to the bare tube.

Table 1. *Advantages and Disadvantages of Air Cooled Heat Exchange Devices*

Advantages
Since water is not used as the cooling medium, the disadvantages of using water are eliminated.
Eliminates high cost of water including expense of treating water.
Thermal or chemical pollution of water resources is avoided.
Installation is simplified due to elimination of coolant water piping.
Location of the air cooled heat exchangers is independent of water supply location.
Maintenance may be reduced due to elimination of water fouling characteristics which could require frequent cleaning of water cooled heat exchangers
Air cooled heat exchangers will continue to operate (but at reduced capacity) due to radiation and natural convection air circulation should a power failure occur.
Temperature control of the process fluid may be accomplished easily through the use of shutters, variable pitch fan blades, variable speed drives, or, in multiple fan installations, by shutting off fans as required.
Disadvantages
Since air has relatively poor thermal transport properties when compared to water, the air cooled heat exchanger could have considerably more heat transfer surface area. A large space requirement may result.
Approach temperature differences between the outlet process fluid temperature and the ambient air temperature are generally in the range of 10 to 15 °K. Normally, water cooled heat exchangers can be designed for closer approaches of 3 to 5 °K. Of course, closer approaches for air cooled heat exchangers can be designed, but generally these are not justified on an economic basis.
Outdoor operation in cold winter environments may require special consideration to prevent freezing of the tube side fluid or formation of ice on the outside surface.
The movement of large volumes of cooling air is accomplished by the rotation of large diameter fan blades rotating at high speeds. As a result, noise due to air turbulence and high fan tip speed is generated.

This bond may be mechanical or metallurgical in nature. Metallurgical bonds are those, in which a solder, braze, or galvanizing alloy coats the fin and bare tube or in which the fin is welded to the tube. Fins, which are extruded or machined from the base tube and are, therefore, integral with the tube, may also be considered as having a metallurgical type bond. Mechanically bonded tubes may be of two types. First, ***imbedded*** or ***grooved*** tubes are formed by machining a helical groove along the length of the tube. The fin is located in the groove and wrapped around the tube, after which the tube material is deformed at the base of the fin. This procedure holds the fin in place and in contact with the tube.

Mechanically bonded tubes may be obtained by mechanically stressing the fin material and/or the tube material to hold the two elements in pressure contact with one another. So called tension wound fins are formed by winding the fin material under tension in a helical manner along the length of the tube.

This method stresses the fin material to maintain contact with the tube. The ends of the fins must be held in place to keep the fins from loosening. This may be done by means of stapling, brazing, soldering, welding or any other way to keep the fins from unwrapping.

Individual fins may be preformed and inserted over the tube, after which the mechanical bond may be obtained by either shrink fitting the fins onto the tube or by expanding the tube radially outward to make pressure contact with the fin material. The means to expand the tube may be hydraulic by pressurizing the tube beyond its yield point; or it may be of a mechanical nature, in which an oversized ball or rod is pushed through the length of the tube, forcing the tube material outward against the fin.

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Tubes whose fins are integral with the tube may also be classified as a mechanical bond type, if a liner tube is used inside the finned tube. A liner tube of another material may be used for compatibility with the tube side process fluid. The contact between the two materials could be formed by expanding the liner tube or by drawing the outer finned tube down over the liner. The operating temperatures of the exchanger, including upset or transient conditions may affect the bonding method, which can be used for the finned tubes. In order to maintain design thermal performance, the bond between the fin and the tube must not deteriorate due to a loosening of the fin, which could result from unequal thermal expansion of the fin and tube materials. In order to avoid this degradation of tube performance, mechanically bonded tubes of the tension type are normally limited to temperatures of 400 to 600 °K; and mechanically bonded grooved fin types from 600 to 700 °K. Metallurgically bonded tubes are limited to temperatures below the melting point of the bonding alloy or to a temperature, dependent upon the physical properties of the tube and fin materials.

The operating environment may influence the choice of materials used and the shape of the fin. Aluminum is very often satisfactory as a fin material, although

copper, steel and stainless steel fins are also used. The fin shape may be of edge-type, L-foot type or double L-foot design. The edge type is used for the grooved fin tube, and in cases, where the base tube is not subject to corrosion.

The L-foot fin covers the tube more or less completely to protect the base tube against corrosive attack, but still leaves a potential corrosive site at the base of the fin adjacent to the preceding fin. The double L-foot is intended to provide complete coverage of the tube, where corrosion would otherwise be a problem. Where corrosion is troublesome, soldered or galvanized tubes may offer a solution. The dimensions of finned tubes are results of past experience in the design of air cooled heat exchangers. Tube diameters range from about 1.905 cm (0.75 in.) to 5.08 cm (2.0 in.).

Helically wrapped fins are fabricated such that the fin height can be between about 3/8 to 3/4 of the tube diameter, but limited because of fabrication requirements to a maximum of about 2.54 cm (1.0 in.) in height. Fin spacings vary between about 275 and 450 fins per meter of tube length, while fin thicknesses range from 0.025 to 0.075 cm. For particular cases these parameters may be varied further.

Tube Bundle

A typical tube bundle arrangement is illustrated in Figure 2. The finned tubes are assembled into the tube bundle. Tube lengths range from about 1.83 m long to as much as 12.2 m long. The number of tube rows deep in the bundle is a function of the performance required and generally ranges between 3 and 30. The ends of the tubes are not finned. This permits the tubes ends to be inserted into tubesheets, located at each end of the bundle. The tubesheets separate the cooling air on the fin side from the process fluid on the tube side. Generally, the tube ends are roller expanded into the tube holes in the tubesheet to form the joint, although for higher pressure applications these may be welded joints.

The tubesheets are attached to tube side headers, which contain the tube side fluid and distribute it to the tubes. The headers may be designed to permit any number of tube side passes for the process fluid. For multipass tube bundles, the headers contain partition plates, which divide the bundle into separate passes. However, these may be limited by the operating temperature conditions. If there is a large temperature difference per pass, then the hotter tubes may expand lengthwise to a much greater extent than the tubes in succeeding passes. This could result in high stresses on the tube joint, resulting in leakage at the joint. If differential expansion between passes is excessive, split headers may be necessary. The tube bundle is normally permitted to float independently of the supporting structure due to overall bundle expansion.

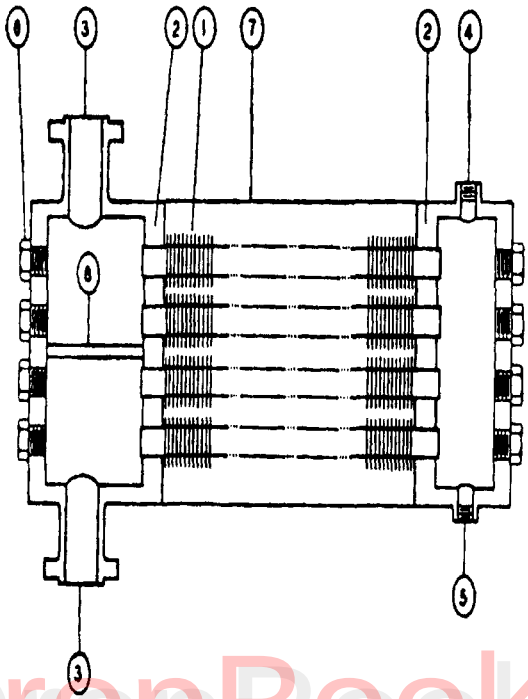

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Figure 2. *Typical tube bundle (two pass) using box headers with tube plugs opposite each tube end. Key: (1) Tube; (2) Tube Sheet; (3) Inlet/Outlet Nozzles; (4) Vent; (5) Drain; (6) Tube Plugs; (7) Side Frame; (8) Pass Rib .*

End plates on the tube side headers frequently include removable plugs. These can be pipe tap plugs or straight threads with gasket seals. The plug are located opposite each tube end to permit access for each tube for re-rolling of the tube to tubesheet joint, should leaks, occur and for cleaning the tubes if this should be necessary. If the tubes are welded into the tubesheets and the process fluid conditions are non-fouling, these plugs are not necessary.

An alternate method of providing access to all tubes for repair and cleaning is to use removable bonnet headers. These designs require gaskets to keep the process fluid from leaking to the atmosphere, but may be advantageous for high tube side fouling conditions. Special header designs may be provided for high tube side pressure conditions. These may be circular headers with individual tubes welded in place or billet type headers with flow passages machined into thick steel sections.

The tube bundle is fabricated as a rigid structure to be handled as an individual assembly. Structural steel side members and tube supports are used for this purpose. Such supports are used beneath the bottom of the tubes to prevent the

bundle from sagging; between tube rows to maintain tube spacing and prevent meshing or deformation of the fins; and across the top row of tubes to keep the tubes in proper position. The supports are spaced evenly along the bundle length at intervals, not exceeding about 1.5 meters.

Fan and Drive Assemblies

Fans are used, which correspond to the dimensions of the tube bundle and the performance requirements for the heat exchanger. Normally, the fan diameter is approximately equal to the bundle width, although smaller diameters may be used. For square, or nearly square bundles, one fan is used. For long rectangular bundles, a number of fans operating in parallel may be used. Fans are of axial flow design, which move relatively large volumes of air at low pressure. In order to minimize air recirculation and improve fan efficiency, fan blades are set within orifice rings which provide close radial clearance between the ring and the blade tips. The ring often has a contoured shape to provide a smooth entrance condition for the air. This minimizes air turbulence at this point, which also helps to reduce noise, generated by the fan.

Rotating at high speeds, the fan blades must be balanced to insure that centrifugal forces are not transmitted through the fan shaft to the drive or to the supporting structure. An unbalanced blade could result in severe vibration conditions. Blades are frequently made of aluminum, but other metals and plastics have also been used. Consideration of maximum operating temperature must be given when using the plastic blades. Where corrosion is possible, blades can be coated with epoxies or other suitable protective material. Smaller diameter fans, up to about 1.5 or 2 meters in diameter, can be driven with electric motors. Larger diameter fans are usually indirectly driven by electric motors or steam turbines, using V-belts or gears. V-belt drives are often limited to fan diameters of about 3 meters and less and motors not exceeding 30 hp.

For larger motors and larger diameter fans, right angle gear drives are used. Indirectly driven fans can offer the advantage of speed variation, such that, as the air cooler heat toad varies, the volume of cooling air can also be varied. The fan laws, which relate speed to fan performance show, that reducing speed can also reduce power consumption. The fan may be designed for either forced air flow or induced air flow. In forced-flow installations, the fan blows ambient air across the tube bundle. Induced-draft fans draw the air across the bundle. Therefore, the fan blades are in contact with the heated air, coming off the heat exchanger. This situation gives a power advantage for the forced draft design.

The total pressure of the fan is the sum of the static pressure loss of the air flowing across the tube bundle, plus the velocity pressure of the air, moving

through the fan. Static pressure losses are of the order of 0.5 cm to 3 cm water gauge, while fans are usually designed for velocity pressure of about 0.25 cm water gauge. The actual volumetric flowrate of air, for a given mass flowrate, is directly proportional to the absolute temperature of the air.

Fan efficiencies are typically about 65 % while drive efficiencies are 95 % or better. This power advantage for forced-draft designs generally proves to result in a more economical heat exchanger. Since the fan is close to the ground, structural costs may be less with the drive assembly, located at ground level. However, induced-draft air cooled heat exchangers offer the advantage of better air distribution across the bundle, due to relatively low air velocities approaching the tubes. Furthermore, the air exit velocities of induced-draft heat exchangers are much higher than a forced-draft design. Thus, the possibility of recirculating hot discharge air is less for the induced-draft. When cooling the process fluid to a temperature close to the inlet ambient air temperature, this may be of particular importance.

In a typical air cooled application, impeller air flow is used to cool media, flowing through the banks of heat exchangers. As in many cases, there is only a single air source, and, hence, the design of a heat exchanger effects the other in the heat exchanger bank. A typical example is a radiator/cooler oil package. As the air flow has to take away heat from the radiator and the oil cooler, both must be designed optimally to make the most efficient package. Any over-designing on any of the units, radiator or oil cooler, will adversely effect the performance of the other.

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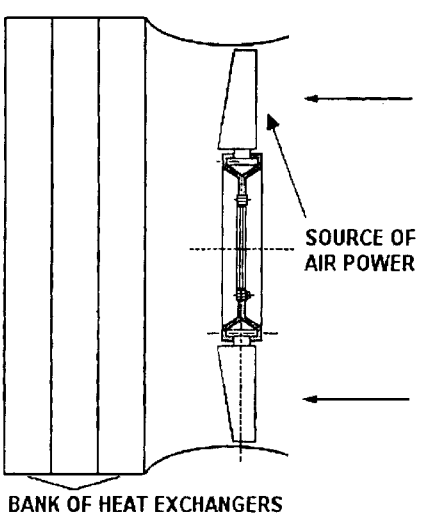

Figure 3. *High-efficiency aerofoil axial impeller.*

As noted earlier, the impeller is a central part of any air cooled heat exchanger. To ensure that the best performance is achieved and power consumption and noise levels are as low as possible, it is important that the proper impeller is selected. Figure 3 shows a multi-wing high-efficiency aerofoil axial impeller. Multi-wing axial impellers can be used in almost any application.

Air Plenum Chamber

The velocity of the air, flowing through the fan, can be as much as 3 to 4 times the velocity across the face of the tube bundle. Also, the air, coming from the circular shape of the fan, must be distributed across the square or rectangular shape of the bundle. The air plenum chamber is intended to make this velocity and shape transition, such that the distribution of air is uniform across the bundle. Common practice is to install the fan in a chamber, such that the distance from the first row of the tube bundle to the fan is about one-half the fan diameter

The plenum chamber design may be a simple box shape, formed by flat sides and bottom, or curved transition sections may be used to obtain a tapered smooth transition from the rectangular bundle to the circular fan. Either design may be used for forced-draft or induced-draft air cooled heat exchangers.

Structural Assembly

The structural assembly of the air cooled heat exchanger is strongly dependent upon the particular plant site requirements of the user. Taken into account should be mechanical loads upon the heat exchanger structure, due to its own weight, of course, but other loadings, such as wind loads, impact loads, nozzles loading and seismic forces must be considered. The presence of equipment beneath the air cooled heat exchangers may require particular designs. Safety considerations may call for fencing or fan guards. Environmental factors could indicate the need for louvers, hail screens, or other protective devices. In addition, the physical location of the heat exchangers may require ladders, platforms, railings, safety cages and other miscellaneous items, which the user will require.

General Exchanger Configurations and Applications

As described above, in the air-cooled exchanger a motor and fan assembly forces ambient air over a series of tubes to cool or condense the process fluids carried within. The tubes are typically assembled in a coiled configuration. Air is

inexpensive and abundant, but it is a relatively poor heat transfer medium. To increase the heat transfer rates of the system, the tubes in air-cooled exchangers are typically given fins, which extend the surface area, increase heat transfer, and give such systems the nickname ***fin-tube coils***. Air-cooled exchangers are typically found in such applications, as heating and air conditioning, process heating and cooling, air-cooled process equipment, energy and solvent recovery, combustion air preheating, and fluegas reheating.

The diameter and materials, specified for the tubes and fins, depend on system requirements. The fins are commonly made from aluminum or copper, but may be fabricated of stainless or carbon steel. Tubes are generally copper, but can be made from almost any material, and they range in size from 5/8- to 1-in. outer diameter. The design of the air-cooled exchanger is such, that individual coils can be removed independently for easy cleaning and maintenance. There are several common design configurations that are commercially available. Each is briefly described below.

Aluminum brazed-fin exchangers consist of corrugated plates and fins, which are added to a brazed-composite core to create alternating air and fluid passages. This compact, lightweight design is considered the most cost-effective air cooled unit available. Turbulence created in the fluid channels boosts efficiency. Typical applications include cooling lube oil for power equipment, cooling fluids for hydraulic equipment, and cooling gear box fluids.

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Aluminum plate-fin exchangers are constructed with traditional heat exchanger tubing. Stacked, die-formed aluminum plates extend the surface to maximize air-side heat transfer. Like the brazed-fin exchanger, this unit is also used for oil and glycol cooling, but its higher flowrate expands its capabilities. Constructed from standard components, aluminum-fin exchangers are designed with a more solid construction than their brazed-fin counterparts. Typical applications include oil cooling, compressed-air cooling, water cooling with air.

Fin-tube exchangers consist of one continuous fin wrapped spirally around a series of individual tubes. Often referred to as a "heavy duty coil", this air exchanger has fin-tube attachments that can be built either to ASME and API standards, or to customer specifications. Often used in air-heating applications, the heavy-duty coil is available with several different fin variations, including the tapered fin, footed 'V' fin, overlapped-footed fin and the embedded fin, which describe the geometries at the fin-tube interface. The method of attaching the fin to the tube is critical, since the loosening of this bond may hinder heat exchange. Typical applications include those that heat air with high-pressure or high-temperature steam, heating or cooling applications with high liquid flows, cannot tolerate condensate freezing - such as steam applications, and heating air with hot water.

The Fine Wire Heat Exchanger

These types of heat exchanger systems are typically used for indoor climate control. On a flat surface, one has a heat transfer coefficient to air of about 20 W/m^2K. On a fine wire, say a wire of 0.1 mm, one can reach 300 W/m^2K. Fine wires have the same cost per square meter as flat surfaces. This type of air cooled heat exchanger generally uses a ceiling fan that can heat or cool air with only a few degrees °C. The fine wire heat exchanger consists of a woven cloth stacked in parallel strips around a slow turning Sirocco-type fan. This type of fan has applications for space cooling and heating, improves the COP (Coefficient of Performance) of heat pumps and makes small-scale seasonal storage possible.

Let's consider the physics of heat transfer to a fine wire. In doing so, a practical set of issues is to determine the optimal wire diameter (100 μ) at which wire cost/performance is minimal, and come at a heat transfer from air to wire cloth as a function of air velocity. A next step would be to treat the pressure drop through wire cloth, and find the optimal air speed (0.4 m/s), at which the sum of the energy loss of pumping and heat transfer is minimal. With these values we find the optimal temperature drop (2.4 °C) over the heat exchanger by minimizing the yearly cost of the exchanger due to investment, and the cost of energy loss due to heat transfer.

A widely used correlation for heat transfer from a cylinder in a perpendicular flow is:

$$Nu = 0.57 \times Re^{0.5} \times Pr^{0.33}$$

$$\text{and } a = Nu \times l/d$$

When we fill in $h = 1.85e\text{-}5$ Pa×s; $\rho = 1.3$ kg/m^3, $C_p = 1010$ J/kg-°K and $\alpha = 0.025$ W/m-°K for the material constants of air at room temperature, this correlation becomes $a = 3.2 \times (v/d)^{0.5}$.

We see that the heat transfer coefficient is inversely proportional to the square root of the wire diameter, which is the reason for the development of fine wire heat exchangers after all. With an air velocity v of 0.5 m/s and a wire of 100 m, we have $a = 226$ W/m^2K, which is around ten times the typical value of flat plate heat exchangers to air.

Fine wires can only be efficiently incorporated into a device using textile technology, such as weaving, and in the case of heat transfer from water to air, one has to weave copper capillaries into copper fine wires. This leads naturally to a cloth where at the capillaries, the wires have a spacing equal to their diameter, and at the wire crossing in the mid point between the capillaries, a zero spacing between the capillaries. This cloth, transferring heat to the air streaming through

it, can be represented as two rows of cylinders, spaced $2\times d$, in series, perpendicular to the flow. Per square meter there are then $1/d$ wires, each with surface $p\times d$ m^2, so that the heat transfer coefficient related to the cloth surface is $a=p\times 3.2\times(v/d)0.5=10\times(v/d)^{0.5}$ W/m^2-°K.

The kilogram price of copper fine wire increases with decreasing wire diameter, because of the wire drawing cost. If P_c is the copper wire price per kg, and the density of copper is 8900 kg/m^3, then per square meter cloth there is $1/d\times p/4\times d^2\times 8900\times P_c$ or $7e3\times d\times P_c$ nlg copper. Per unit of heat transfer coefficient, $10\times(v/d)^{0.5}$ W/m^2-°K, there is $700\times d^{1.5}\times v^{-0.5}\times P_c$ nlg copper. So we have the economically optimal wire diameter, when the product $Pe\times d^{1.5}$ is minimal. This is the case with a wire diameter of 100 mm. With this optimal wire diameter, the calculated heat transfer coefficient, related to the cloth, surface is $1000\times v^{0.5}$ W/m^2-°K.

The following correlation for pressure drop describes flow perpendicular to pipe bundles for $Re < 25\times x/(x-1)$, where x is the ratio of pipe spacing to pipe diameter, v means the mean fluid velocity over the bundle front surface, l - the pipe bundle length in the flow direction, and d - the pipe diameter. In our case, we have two bundles in series with $x=2$ and with $l/d=1$ at a distance of more than $2\times d$, and $d=d=1e\text{-}4$ m, so we can write here for the pressure drop:

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$$\Delta P=2\times 28.4/p\times h/r/v/d\times 1\times 0.5\times r\times v^2 ;$$

or $\Delta P= 6.6\times v$, for $Re<25$ or $v<3.8$ m/s

By blowing air through the cloth we have to use energy, or electric energy. When the ceiling ventilator motor has a constant efficiency of $6W_{mech}$ / 45 W_{el}, and the total pressure drop, inclusive the acceleration term, through the fan $\Delta P=6.6\times v+0.5\times v^2$, the surface of the heat exchanger is A, then we need $45/6\times\Delta P\times A$ electric power to move the air.

By moving the air faster, we increase the heat transfer, and so save energy by lowering the temperature drop through which the heat flows. We can express this energy by estimating the electric energy, we need to pump heat from outside air at a mean temperature over an arbitrary heating season of 4.8 °C to heat inside air at a mean temperature of 17.3 °C.

For each increase of the delta temperature of one of the heat exchangers by 1 °C, we need an extra F_w /(17.3-4.8)/ 8 amount of electric power when the COP of a heat pump is 8. When we use the heat transfer and pressure drop equations in these functions for a heat exchanger of $F_w=1$kW, the sum of the energy losses is:

$$45/6 \times A \times (0.5 \times v^2 + 6.6 \times v) + (1kW)^2 / 8/12.5/ 1000 \times v^{0.5} \times A, \text{ or}$$

$$A \times (3.75 \times v^2 + 49.5 \times v + 10 \times v^{-0.5})$$

The value of v, at which this function has its minimum value, is 0.21 m/s. In practice, the efficiency of a standard one-phase fan motor increases with its output, and this increases the optimum air velocity to about v = 0.4 m/s in a typical case. This low optimal air speed is the reason that the heat exchanger surface should be plied in a zig-zag fashion to increase the frontal air speed to a more acceptable value of about 2 m/s, in order to keep the apparatus in a compact form.

When we take v to be 0.4 m/s, our cloth surface at 1 kW becomes

$$A = 1e^3 / 1000.0.4^{0.5}/\Delta T = 1.6 / \Delta T \ m^2(nlg),$$

or when we write the cloth cost of Pa off in 7 years, $1.6/7 \times Pa/\Delta T$ nlg/y.

The energy loss is $1kW \times \Delta T/12.5/COP = 10 \times \Delta T$ (W), this power during the heating season of 5080 hours, $10 \times \Delta T \times 5080 = \Delta T \times 50.8$ kWh/y, or with a kWh price at 0.2 nlg, $10.2 \times \Delta T$ nlg/y. The sum of capital cost and energy loss cost is $0.23 \times Pa/DT + 10.2 \times \Delta T$. With Pa = 300 nlg/m^2 cloth, mounted in a heat exchanger, this function is minimal with Pa = 300 nlg/m^2, at T=2.6 °C, and with Pa = 200 nlg/m^2, at ΔT= 2.2 °C. The cost price is one of the last parameters that becomes known, but a value for the optimal ΔT = 2.5 °C can be taken. This in sharp contrast with usual tube-plate-fin heat exchangers, that have optimal temperature drops of 15 to 20 °C.

In home or building heating applications, a slowly turning fan drives air (typically 500 liters per second) through a maze of copper fine wire gauze, woven around water carrying copper capillaries: a fine wire water-to-air heat exchanger. The result is one million 100μ, 0.6 mm long copper fins that transfer heat ten times better than a typical central heating radiator. The performance can be described with the following formulae in Table 2, where D = diameter fan in m; n = speed of fan in RPM.

Table 2. *Formulae for Fine Wire Heat Exchanger*

Parameter	Formula	Units
Heat transfer	$8 \times D^2 \times n^{0.8}$	W/°C
Fan power (Pm)	$3 \times 10^{-5} \times D^4 \times n^3$	W
Noise level	$20 \times \log Pm + 28$	dB(A)
Cost in series of 1000	$280 \times D$	U.S. $

A fan with a diameter of 60 cm and a speed of 100 RPM will have a heat transfer capacity of 100W/°C, needs a mechanical power of 4W, has a noise level of 40 dB(A) and costs 180 Euro.

Using this type of heat exchanger fan, rooms and working spaces can be cooled with ground water, that is only about 3 to 5 degrees cooler, than the desired room temperature. A leading European manufacturer of these types of heat exchanger systems is Fiwihex in The Netherlands, where ground water is normally about 12 °C. Hence, one 100 W/°C ceiling fan can cool a room to 20 °C with a power of (23-12)×100 = 1100 Watt. This is about the power (3750 BTU/hr) that a portable air conditioner can deliver. As an example, when we need this 100 W/°C = $8 \times D^2 \times n^{0.8}$, select a fan with a D of 60 cm. The speed n becomes then $(2000/(20-12)/8/0.6^2)^{(1/0.8)}$ = 84 rpm, the mechanical power becomes $3 \times 10^{-5} \times 0.6^4 \times 84^3$ = 2.3 Watt, and the noise level becomes 20 log 2.3 + 28 = 35 dBA. This is a level of noise that is so low that it is never reached inside a city environment. The fan and water circulation pump will use about 60W of electricity, this has to be compared with 400W for a typical air conditioner.

The heat exchangers described are so powerful that heat extraction from the outside air is now an economic possibility, when no ground or surface water is available. A Coefficient Of Performance of 6 is attainable as a mean over the Dutch heating season (inside temperature 20 °C, outside 4.8 °C) according to Fiwihex, with the following configuration: 2×1000 W/°C propane-to water heat exchangers; 2×500 W/°C fans and a standard shop display refrigeration compressor. The heat pump stops during electricity peak hours when the heat storage system has been installed. When (ground)water as a heat source is available, the COP rises to 8.

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SHELL AND TUBE TYPE HEAT EXCHANGERS

Tubular Exchanger Manufacturers Association (TEMA)

The Tubular Exchanger Manufacturers Association, or TEMA, is a group of leading manufacturers, who have pioneered the research and development of heat exchangers for over fifty-five years. Founded in 1939, TEMA has grown to include a select group of member companies. Although it may be easy to choose a TEMA member as a supplier, it is not easy for manufacturers to become a TEMA member. Member companies must meet stringent criteria to even qualify for TEMA membership, and are periodically examined by TEMA to ensure that the manufacturer meets membership criteria, and designs and manufacturers according to TEMA standards. Members adhere to strict specifications. TEMA Standards and Software have achieved worldwide acceptance as the authority on shell and tube heat exchanger mechanical design. These tools give engineers a

valuable edge when designing and manufacturing all types of heat exchangers. Seven editions of TEMA Standards have been published, each on updating the industry on the latest developments in technology. TEMA has also developed engineering software that complements the TEMA Standards in the areas of flexible shell elements (expansion joints) analysis, flow induced vibration analysis and fixed tubesheet design and analysis. This state-of-the-art software works on an IBM PC or compatible, and features a materials data-bank of 38 materials, as well as user-friendly, interactive input and output screens. The programs handle many complex calculations, so users can focus on the final results. Many companies can manufacture heat transfer equipment, but not all can assure its safe, effective design and quality construction. That's why the TEMA Heat Exchanger Registration System was instituted in 1994. For quality assurance, one need only look for the TEMA Registration Plate attached to the heat exchanger. Each plate includes a unique TEMA registration number. Before a company can even become a member of TEMA and participate in the registration system, it must have a minimum of 5 years of continuous service in the manufacture, design and marketing of shell and tube heat exchangers.

All TEMA companies must have in-house thermal and mechanical design capabilities, and thoroughly understand current code requirements and initiate strict quality control procedures. Additionally, all welding must be done by the company's own personnel, and the company must have its own quality control inspectors. These criteria ensure the highest level of technical expertise, which gives TEMA members a meaningful advantage when designing or fabricating heat exchangers. The following is a list of TEMA manufacturers.

Alco Products - 8505 Jacksboro Highway, Wichita Falls, TX 76302-9703 (Phone: 940-723-6366 Fax: 940-723-1360)

API Heat Transfer Inc. - 2777 Walden Ave, Buffalo, NY 14225 (Phone: 716-684-6700 Fax: 716-684-2129)

Cust-O-Fab, Inc. - 8888 W 21st St, Sand Springs, OK 74063 (Phone: 918-245-6685 Fax: 918-241-1434)

Energy Exchanger Co. - 1844 North Garnett Road, Tulsa., OK 74116 (Phone: 918-437-3000 Fax: 918-437-7144)

Engineers & Fabricators Co. - 3501 West 11th Street, Houston, TX 77008-6001 (Phone: 713-803-4700 Fax: 713-869-8088)

Fabsco Shell & Tube, LLC - P.O. Box 988, Sapulpa, OK 74066 (Phone: 918-224-7550 Fax: 918-224-3564)

Graham Corporation - 20 Florence Avenue, Batavia, NY 14020 (Phone: 716-343-2216 Fax: 716-343-1097)

Heat Transfer Equipment Co. - P.O. Box 580638, Tulsa, OK 74158 (Phone: 918-836-8721 Fax: 918-838-3570)

Hughes-Anderson Heat Exchangers, Inc. - 1001 N. Fulton Ave, Tulsa, OK 74115 (Phone: 918-836-1681 Fax: 918-836-5967)

ITT Standard - 175 Standard Parkway, Buffalo, NY 14240 (Phone: 716-897-2800 Fax: 716-897-1777)
Joseph Oat Corporation - 2500 Broadway, Camden, NJ 08104 (Phone: 856-541-2900 Fax: 856-541-0864)
Manning & Lewis Eng. Co. - 675 Rahway Ave, Union, NJ O7083 (Phone: 908-687-2400 Fax: 908-687-2404)
Nooter Corporation - P.O. Box 451, St, Louis, MO 63166 (Phone: 314-421-7561 Fax: 314-421-7580)
Ohmstede, Inc. - St. Gabriel, Louisiana 70776 (Phone: 409-833-6375 Fax: 409-833-6735)
R.A.S. Process Equipment, Inc. - 324 Meadowbrook Road, Robbinsville, NJ 08691 (Phone: 609-371-1000 Fax: 609-371-1221)
Southern Heat Exchanger, Corp. - P.O. Box 1850, Tuscaloosa., AL 35403 (Phone: 205-345-5335 Fax 205-345-0009)
Steeltek, Inc. - 4141 South Jackson, Tulsa, OK 74107 (Phone: 918-446-4001 Fax: 918-446-8317)
Struthers Industries,Inc. - 1500 34th Street, Gulfport, MS 39502 (Phone: 228-864-5410 Fax: 228-864-5555)
Wiegmann & Rose - (Xchanger Manufacturing Corp.) P.O. Box 4187, Oakland, CA 94614 (Phone: 510-632-8828 Fax: 510-632-8920)
Yuba Heat Transfer - (div. of Connell Ltd, Partnership) P.O. Box 3158, Tulsa, OK 74101 (Phone: 918-234-6000 Fax: 918-234-3345)

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The greatly expanded 8th Edition of the Standards of the Tubular Exchanger Manufacturers Association retains the useful data and features, found in the Seventh Edition, plus many clarifications and innovations. All sections have been reviewed to incorporate new data, which were not available at the time of the 1988 printing, including suggestions, which resulted from the extensive use of the Standards by both manufacturers and users of shell and tube heat exchangers. Many helpful recommendations were also received through the cooperation of the American Petroleum Institute (API) and the American Society of Mechanical Engineers (ASME). Some noteworthy features of the Eighth Edition include: **(a)** Metrification has been included where feasible and appropriate; **(b)** Methods for calculating several types of floating head backing rings have been added; **(c)** A method for incorporating pass partition rib area into flange design has been incorporated; **(d)** The vibration section has been expanded and vibration amplitude for vortex shedding and acoustic resonance have been added; **(e)** Nozzle flange pressure/temperature rating tables from ASME Standard B16.5-1996 w/ 1998 addenda are included; **(f)** New materials have been included in coefficient of thermal expansion, modulus of elasticity, and thermal conductivity tables; **(g)** Design equations for double tubesheets have been added; **(h)** A method for calculating the mean metal temperature for tubesheets has been added; **(i)** Stress multipliers have been added to account for the stiffness of

knuckles on flanged and flued expansion joints; **(j)** Suggested calculation methods have been incorporated for both vertical and horizontal supports; **(k)** Design methods have been added for lifting lugs; **(l)** A demonstration copy of the available software is included with the purchase of each Standard.

The Tubular Exchanger Manufacturers Assn. has established heat exchanger standards and nomenclature. Every shell-and-tube device has a three-letter designation; the letters refer to the specific type of stationary head at the front end, the shell type, and the rear-end head type, respectively (a fully illustrated description can be found in the TEMA standards). Common TEMA designations are listed with specific configurations described below.

Shell and Tube Configurations

The shell and tube heat exchanger consists of a shell, usually a circular cylinder, with a large number of tubes, attached to an end plate and arranged in a fashion where two fluids can exchange heat without the fluids, coming in contact with one another. The most common types of heat exchangers configurations are illustrated in Figure 4.

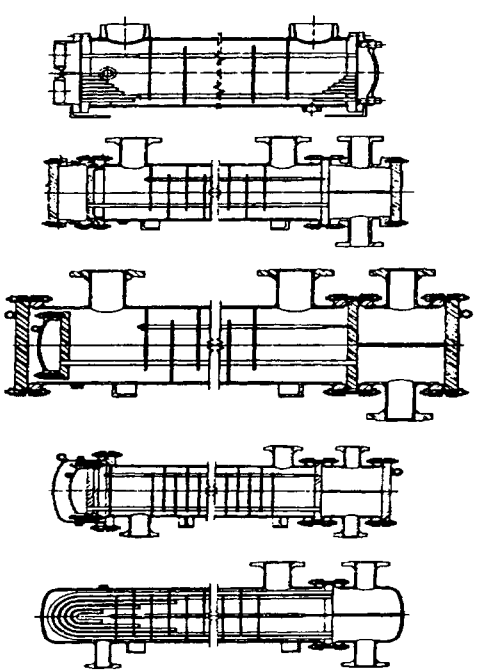

Figure 4. *Common shell-and-tube exchanger configurations.*

There are many text books that describe the fundamental heat transfer relationships, but few discuss the complicated shell side characteristics. On the shell side of a shell and tube heat exchanger, the fluid flows across the outside of the tubes in complex patterns. Baffles are utilized to direct the fluid through the tube bundle and are designed and strategically placed to optimize heat transfer and minimize pressure drop.

A measure of the complexity of predicting shell side heat transfer can be obtained by considering the path of shell side fluid flow. The flow is partially perpendicular and partially paralleled to the tubes. It reverses direction as it travels around the baffle tips and the flow regime is governed by tube spacing, baffle spacing and leakage flow paths. Throughout the fluid path, there are a number of obstacles and configurations, which cause high localized velocities.

These high velocities occur at the bundle entrance and exit areas, in the baffle windows, through pass lanes and in the vicinity of tie rods, which secure the baffles in their proper position. In conjunction with this, the shell side fluid generally will take the path of least resistance and will travel at a greater velocity in the free areas or by-pass lanes, than it will through the bundle proper, where the tubes are on a closely spaced pitch. All factors considered, it appears a formidable task to accurately predict heat transfer characteristics of a shell and tube exchanger.

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The problem is further complicated by the manufacturing tolerances or clearances that are specified to allow assembly and disassembly of the heat exchanger. It is improbable that these clearances will all accumulate to either the positive or negative side, so it is customary to compute heat transfer relationships on the basis of average clearances.

The various paths of fluid flow through the shell side of a segmental baffle heat exchanger is illustrated in Figure 5, whereby the letter designations in the figure are: (A) leakage stream through the annular spaces between tubes and baffle holes of one baffle; (B) cross flow stream through the heat transfer surface between successive baffle windows. It will be noted that this stream is made of B_1 (a portion of fluid passing through baffle windows) plus portions of the A stream; (C) by-pass stream on one side of tube nest flowing between successive baffle windows; (D) leakage stream between shell and edge of one baffle. The by-pass area C between the bundle and shell can be reduced by using dummy tubes, seating strips, or tie rods with seal strip baffles.

The dummy tubes do not pass through the tubesheets, and can be located close to the inside of the shell. The seating strips extend from baffle to baffle in a longitudinal direction and effectively channel the fluid across the tubes to minimize turbulence and heat transfer. On some fixed tubesheet designs, the outer tubes are in close proximity to the inside of the shell so that by-pass is minimal and no by-pass elimination is necessary. There are a number of

techniques that can be employed to reduce the flow in areas A and E. Tight tolerances are often employed and some manufacturers use a punched collar baffle where the tube holes in the baffle have a small precision collar which minimizes clearances between tube and tubehole with the added benefit of good tube support. The baffles are sometimes welded at the shell's periphery to completely eliminate by-pass. Each of these techniques is effective, but are governed by the trade off of increased efficiency versus added cost.

Shell and tube heat exchangers are generally designed with a certain degree of conservatism from both the thermal and mechanical design aspects. From a thermal design viewpoint, the conservatism arises from excessive surface to accommodate fouling in service. From a mechanical design viewpoint, design procedures generally employ allowable stresses, which are based on a factor of safety. But, even so, shell and tube heat exchangers experience problems in service. One of these problems concerns fouling of either the tube side or shell side of the heat exchanger.

Fouling is an accumulation of scale or dirt on the tube surface, thereby adding a resistance to heat transfer. It is very difficult to accurately predict the degree of fouling for a specified service period. There are minimal documented test results on this subject and the results are seldom applicable because of the number of variables in a fouling study. It is, indeed, a fortunate user, who can rely on past performance of the same or similar equipment and specify the proper amount of excess surface required to offset the amount of fouling. For most applications, the degree of fouling is strictly an estimate and the probability is that the heat exchanger is either inadequate or over surfaced. Once the tubes are fouled, they can be either mechanically or chemically cleaned.

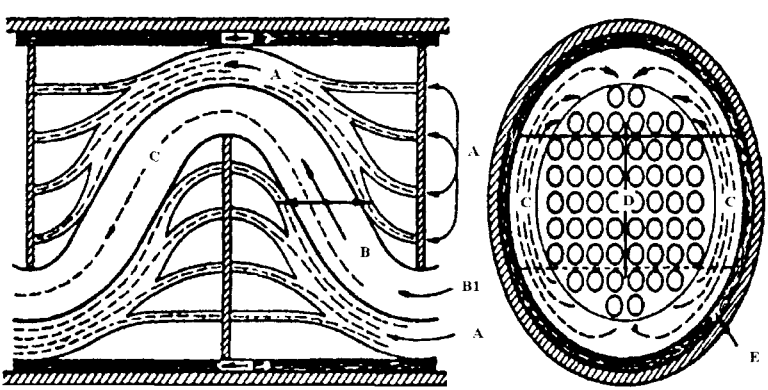

Figure 5. *Illustrates leakage path streams.*

Generally, the tube side presents no particular problem and straight tubes can be easily wire brushed. U-tubes are difficult to clean mechanically and are generally used, where fouling is expected to be minimal. The shell side of the heat exchanger is more difficult to clean, particularly for closely spaced staggered types of tube bundles. Many users specify square or rectangular pitch tube arrangement and removable bundle construction where excessive shell side fouling is expected.

Another serious problem in heat exchangers is corrosion. Severe corrosion can and does occur in tubing and very often with common fluids such as water. Proper material selection based on a full analysis of the operating fluids, velocities and temperatures is mandatory. Very often, heavier gauge tubing is specified to offset the effects of corrosion, but this is only a partial solution. This should be followed by proper start-up, operating and shut-down procedures.

Many heat exchangers use water on the tube side as the cooling medium and compatible copper alloy tubing and still experience corrosion problems. Invariably, this can be traced to some part of the cycle, where the water was stagnant or circulated at extremely low velocity. Most problems with heat exchangers occur during initial installation or shortly thereafter. Improper installation or misalignment can create excessive stresses in supports or nozzles or cause damage to expansion joints or packed joints.

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On initial start-up and shut-down the heat exchanger can be subjected to damaging thermal shock, overpressure or hydraulic hammer. This can lead to leaky tube-to-tubesheet joints, damaged expansion joints or packing glands because of excessive axial thermal, expansion of the tubes or shell. Excessive shell side flowrates during the "shake down" can cause tube vibrations and catastrophic failure.

Table 3 provides recommended start-up and shut-down procedures. Effort should be made to avoid subjecting units to thermal shock, overpressure, and/or hydraulic hammer, since these conditions may impose stresses, that exceed the mechanical strength of the unit or the system in which it is installed, which may result in leaks and/or other damage to the unit or entire system.

Some general considerations to bear in mind are: (1) In all start-up and shut-down operations, fluid flows should be regulated so as to avoid thermal shocking the unit, regardless of whether the unit is of either a removable or non-removable type of construction; (2) For fixed tubesheet (i.e., non-removable bundle) type units, where the tube side fluid cannot be shut down, it is recommended that both a bypass arrangement be incorporated in the system, and the tube side fluid be bypassed before the shell side fluid is shut down; (3) Extreme caution should be taken on insulated units where fluid flows are terminated and then restarted. Since the metal parts could remain at high temperatures for extended periods of time, severe thermal shock could occur.

Table 3. *Recommended General Start-Up and Shut-Down Procedures*

Fluid Location and Relative Temp.				Start-Up Procedure	Shut-Down Procedure
Shell Side		Tube Side			
Fluid	Temp.	Fluid	Temp.		
Fixed Tubesheet (Non-Removable Bundle)					
L	H	L	C	Start both fluids gradually at the same time.	Shut down both fluids gradually at same time.
Cond. Gas	H	L or G	C	Start hot fluid first, then slowly start cold fluid. Avoid temperature shock.	Shut down cold fluid first, then hot fluid.
G	H	L	C	Start cold fluid first, then hot fluid.	Shut down cold fluid gradually, then hot fluid.
L	C	L	H	Start both flows gradually at the same time.	Shut down both fluids gradually at same time.
L	C	G	H	Start cold fluid first, then hot fluid.	Shut down hot fluid first, then cold fluid.
U-Tube/ Packed Floating Head/ Packed Floating Tubesheet/ Internal Floating Head (All these types have removable bundles)					
L	H	L	C	Start cold fluid first, then gradually start hot fluid.	Shut down hot fluid first, then cold fluid.
Cond. Gas	H	L or G	C	Start cold fluid first, then gradually start hot fluid.	Shut down cold fluid first, then gradually shut down hot fluid.
G	H	L	C	Start cold fluid first, then gradually start hot fluid.	Shut down hot fluid first, then cold fluid.
L	C	L	H	Start cold fluid first, then gradually start hot fluid.	Shut down hot fluid first, then cold fluid.
L	C	G	H	Start cold fluid first, then gradually start hot fluid.	Shut down hot fluid first, the cold fluid.

C - Cold; H - Hot; G - Gas; L- Liquid; Cond. Gas - Condensing Gas (e.g., steam)

The shell-and-tube exchanger's flexible design, high pressure and temperature capabilities, and its ability to handle high levels of particulate material make it the most common heat exchanger used in the CPI. Mechanically simple in design and relatively unchanged for more than 60 years, the shell-and-tube offers a low-cost method of heat exchange for many process operations. The most common shell-and-tube configurations are briefly described.

Straight-Tube, Fixed-Tubesheet Exchangers

The fixed-tubesheet exchanger is the most common, and generally has the lowest capital cost per ft^2 of heat-transfer surface area. Fixed-tubesheet exchangers consist of a series of straight tubes sealed between flat, perforated metal tubesheets.

Because there are neither flanges, nor packed or gasketed joints inside the shell, potential leak points are eliminated, making the design suitable for higher-pressure or potentially lethal/toxic service. However, because the tube bundle cannot be removed, the shellside of the exchanger (outside the tubes) can only be cleaned by chemical means.

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The inside surfaces of the individual tubes can be cleaned mechanically, after the channel covers have been removed. The fixed-tubesheet exchanger is limited to applications where the shellside fluid is non-fouling; fouling fluids must be routed through the tubes. Common TEMA designations for the straight-tube, fixed-tubesheet exchangers are BEM, AEM, NEN. Common applications include vapor condensers, liquid-liquid exchangers, reboilers, and gas coolers.

Removable-Bundle, Externally Sealed, Floating-Head Exchanger

Floating-head exchangers are so named because they have one tubesheet that is fixed relative to the shell, and another that is attached to the tubes, but not to the shell, so it is allowed to "float" within the shell. Unlike fixed-tubesheet designs, whose dimensions are fixed at a given dimension relative to the shell wall, floating-head exchangers are able to compensate for differential expansion and contraction between the shell and the tubes. Since the entire tube bundle can be removed, maintenance is easy and relatively inexpensive. The shellside surface can be cleaned by either steam or mechanical means. In addition to accommodating differential expansion between the shell and tubes, the floating tubesheet keeps shellside and tubeside process fluids from intermixing. Although the externally sealed, floating-head design is less costly than the full, internal-

floating-head exchanger, it has some design limitations: both shellside and tubeside fluids must be non-volatile or non-toxic, and the tubeside arrangements are limited to one or two passes. In addition, the packing used in this exchanger limits design pressure and temperature to 300 psig and 300 °F. Common TEMA designations are AEW and BEW. Applications include exchangers handling inter- and after-coolers, oil coolers, and jacket water coolers.

Removable- Bundle, Internal-Clamp-Ring, Floating-Head Exchanger

This design is useful for applications where high-fouling fluids require frequent inspection and cleaning. Because the exchanger allows for differential thermal expansion between the shell and tubes, it readily accommodates large temperature differentials between the shellside and the tubeside fluids. This design has added versatility, however, since multi-pass arrangements are possible. However, since the shell cover, clamp ring, and floating-head cover must be removed before the tube bundle can be removed, service and maintenance costs are higher than in "pull through" designs (discussed below). Common TEMA designations are AES and BES. Typical applications include process-plant condensers; inter- and after-cooler designs, gas coolers and heaters, and general-purpose industrial heat exchangers.

Removable-Bundle, Outside-Packed, Floating-Head Exchanger

This design is well suited for applications where corrosive liquids, gases, or vapors are circulated through the tubes, and for air, gases, or vapors in the shell. Its design also allows for easy inspection, cleaning, and tube replacement, and provides large bundle entrance areas without the need for domes or vapor belts. Only shellside fluids are exposed to packing, allowing high-pressure, volatile, or toxic fluids to be used inside the tubes. The packing in the head does, however, limit design pressure and temperatures. Common TEMA designations are BEP and AEP. Common applications include oxygen coolers, volatile or toxic fluids handling, and gas processing.

Removable-Bundle, Pull-Through, Floating-Head Exchangers

In the pull-through, floating-head design, the floating-head cover is bolted directly to the floating tubesheet. This allows the bundle to be removed from the shell without removing the shell or floating-head covers, which eases inspection

and maintenance. This is ideal for applications that require frequent cleaning. However, it is among the most expensive designs. The pull-through design accommodates a smaller number of tubes in a given shell diameter, so it offers less surface area than other removable-bundle exchangers.

Common TEMA designations are AET and BET, and typical applications include exchangers handling chemical fluids, hydrocarbon fluid condensers, air or gas compressors, and inter-and after-coolers.

Removable-Bundle, U-Tube Exchangers

In the U-tube exchanger, a bundle of nested tubes, each bent in a series of concentrically tighter U-shapes, is attached to a single tubesheet, as illustrated in Figure 7. Each tube is free to move relative to the shell, and relative to one another, so the design is ideal for situations that accommodate large differential temperatures between the shellside and the tubeside fluids during service. Such flexibility makes the U-tube exchanger ideal for applications that are prone to thermal shock or intermittent service.

As with other removable-bundle exchangers, the inside of the shell, and to the outside of the tubes. However, unlike the straight-tube exchanger, whose tube internals can be mechanically cleaned, there is no way to physically access the U-bend region inside each tube, so chemical methods are required for tubeside maintenance.

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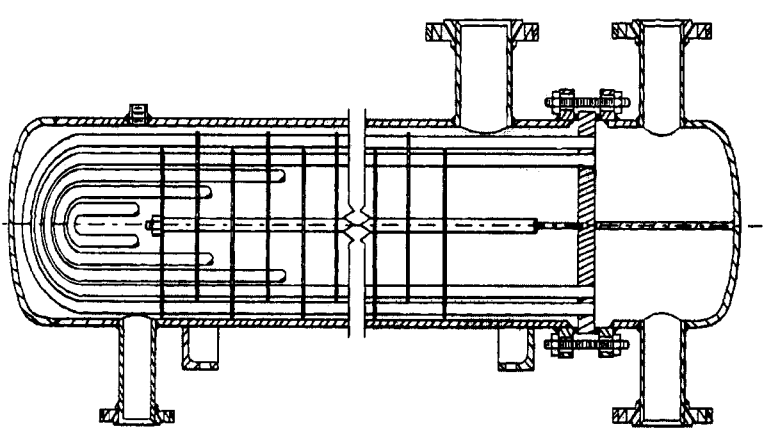

Figure 7. U-tube design.

As a rule of thumb, non- fouling fluids should be routed through the tubes, while fouling fluids should be reserved for shellside duty. This inexpensive exchanger allows for multi-tube pass arrangements. However, because the U-tube cannot be made single pass on the tubeside, true countercurrent flow is not possible. Common TEMA designations are BEU and AEU, and typical applications include oil cooling, chemical condensing, and steam heating applications.

Special Designs

For applications with high vapor-flow and high- pressure conditions, a specially designed shell-and-tube exchanger must often be employed. Special designs may also be called for when applications have close temperature crossings, meaning the outlet temperature of the warmed fluid exceeds that of the cooled fluid.

The following are several examples: TEMA K-type shells, which allow for proper liquid disengagement for reboilers; TEMA J-type shells, which accommodate high vapor flows by allowing for divided flow in the shellside; Two-pass TEMA F-type shells, which can be used for applications when a temperature cross exists (below); TEMA D-type front head designs, which are often the answer for high-pressure tubeside applications.

While these specially designed exchangers may be the solution to a process problem, construction costs tend to be higher than those of "standard" engineered shell-and-tube equipment. Common TEMA designations include BKU, BJM, BFM and DED. Specially designed exchangers are often called for in such applications as reboilers, steam heaters, vapor condensers, and feedwater heaters.

Off-the-Shelf Exchangers

Fixed-tubesheet and U-tube shell-and-tube exchangers are the most common types of off-the-shelf heat exchangers available. Such stock models are typically used as components in vapor condensers, liquid-liquid exchangers, reboilers and gas coolers. Standard fixed-tubesheet units, the most common shell-and-tube heat exchangers, range in size from 2 to 8 in. dia. Materials of construction include brass or copper, carbon steel, and stainless steel. Even though this exchanger is one of the least expensive available, it is still generally constructed to standards specified by the manufacturer. Stock exchangers can be constructed to American Society of Mechanical Engineers (ASME) codes.

U-tube heat exchangers are commonly used in steam heating applications, or heating and cooling applications that handle chemical fluids as opposed to water.

While the U-tube is generally the lowest-priced heat exchanger available, higher service and maintenance costs tend to be higher than other exchangers, since the nested, U-bend design makes individual tube replacement difficult. Custom-designed heat exchangers, though more expensive than their off-the-shelf counterparts, are generally made to higher design standards than stock exchangers.

Many manufacturers follow the TEMA standards for design, fabrication and material selection. TEMA B is the most common TEMA designation, and provides design specifications for exchangers used in chemical process service. TEMA C guidelines provide specifications for units used in commercial and general process applications. TEMA R guidelines provide specifications for exchangers used in petroleum refining and related process operations. Each of these classes are applicable to shell-and-tube heat exchangers with the following limitations: (1) Shell diameter does not exceed 60 in.; (2) Pressure does not exceed 3,000 psi; (3) The product shell diameter (in.) times pressure (psi) does not exceed 60,000. Standards set by the American Petroleum Institute (API; Washington, DC) are also generally accepted throughout the heat exchanger industry. These standards (API 614,660 and 661) specify the mechanical design of the exchanger and list specific materials that can be used in construction of both water-and air-cooled exchangers. While there are significant advantages to purchasing a custom-designed exchanger that meets either TEMA or API manufacturing guidelines, these specifications add to the cost of the exchanger and may slow delivery time.

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SPIRAL-PLATE HEAT EXCHANGERS

A spiral-plate exchanger is fabricated from two relatively long strips of plate, which are spaced apart and wound around an open, split center to form a pair of concentric spiral passages. Spacing is maintained uniformly along the length of the spiral by spacer studs welded to the plate.

For most services, both fluid-flow channels are closed by welding alternate channels at both sides of the spiral plate (Figure 8). In some applications, one of the channels is left completely open on both ends and the other closed at both sides of the plate (Figure 9). These two types of construction prevent the fluids from mixing.

Spiral-plate exchangers are fabricated from any material that can be cold worked and welded. Materials commonly used include: carbo steel, stainless steel, nickel and nickel alloys, titanium, Hastelloys, and copper alloys. Baked phenolic-resin coatings are sometimes applied. Electrodes can also be wound into the assembly to anodically protect surfaces against corrosion.

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Chapter 2
EVAPORATIVE COOLING EQUIPMENT

INTRODUCTION

This chapter discusses an important piece of equipment used throughout many industry sectors, including the CPI, namely cooling towers. Cooling towers are the most basic type of evaporative cooling equipment used primarily for process water cooling purposes. The following are organizations and associations that the reader can contact for further information on cooling towers and related subjects. The Cooling Tower Institute, CTI, is a non-profit organization based in Houston, Texas comprised of cooling tower users, manufacturers, and related service providers. It is probably best known for its test specifications and extensive library of information on cooling tower related subjects. The American Society of Heating, Refrigeration and Air Conditioning Engineers, ASHRAE, is an international organization which is also non profit and headquartered in Atlanta, Georgia. ASHRAE promotes standards based on extensive research, and they publish comprehensive books on the subject. Most of the weather data used by system designers comes from ASHRAE publications. They have numerous local chapters which meet regularly for educational and social events. The Refrigerating Engineers and Technicians Association, RETA, is a non profit organization based in Chicago. They publish educational materials that focus on industrial refrigeration and have numerous local chapters that meet regularly. Users and manufacturers of Evaporative Condensers should be interested in joining. The Association of Energy Engineers is similar to the above with local chapters that deal with energy issues. Cooling Towers are a common fixture in most power plants and therefore receive considerable attention from this group.

THERMAL CHARACTERISTICS

Before discussing actual equipment, a review of the thermal characteristics of evaporative cooling is presented. The latent heat of vaporization has long been used to transfer heat to the atmosphere. Our own bodies, in fact are an example of evaporative cooling, whereby our sweat glands spread water over our skin, and hence, our bodies benefit from the cooling effect that occurs when the liquid evaporates into vapor. Consider the assignment of taking a bucket of water and changing the water to vapor as quickly as possible. Several strategies can be

implemented. One approach might be to use a second empty bucket and pour the water back and forth to expose more of the water's surface area to air. The falling water could be made to splash into droplets to increase the surface area exposed to the air even further. Even more effective would be to take the bucket and dump out the water onto a large surface for maximum exposure of the water to air. To improve the operation further, a fan could be added to pass air over the water to encourage moisture laden air to exit and be replaced with new air that is better able to accept more vapor. Finally, if the quality of air could be manipulated, dry air- air that contains very little moisture- would be used because of its ability to accept greater amounts of water vapor. If the process described were modified such that new water was continually added to replace that lost to evaporation, and the water was continuously recirculated over the surface, and the runoff was continuously heated... Then we would have all the basic steps of an operational cooling system. For each pound of water that a cooling tower evaporates, it removes somewhere around 1000 Btu from the water that remains. The more evaporation that takes place, the more heat that is removed. Additional heat is taken away by the air by virtue of its temperature increase but this sensible heat exchange is minor compared to the latent component provided by the water's phase change.

In a cooling tower system, design criteria are based upon knowledge of the water flow rate, water inlet temperature, water outlet temperature, and the ambient wet bulb temperatures. Wet bulb temperature is a site condition measured by constructing what amounts to a tiny hand-held cooling tower. This small cooling tower has no heat input and is used to determine the lowest leaving water temperature a cooling tower could possibly attain accurately predicting the performance of a larger, operational counterpart. This tiny cooling tower or instrument is called a **sling psychrometer**. It places a thin film of water on the bulb of a thermometer. The thermometer is twirled in the air. After a few seconds, the thermometer begins to show a reduced temperature reading. Twirling it more will yield successively lower temperature readings until a final low temperature reading can be made after about one minute. Additional twirling serves no additional benefit. This low reading is called the **wet bulb temperature**. It is necessary to insure that the thin water film be maintained. A cotton sock connected to a small water reservoir is typically employed. Some psychrometers use a small battery operated fan so that the operator doesn't have to twirl the device in the air. Both types also have a non wetted thermometer that reads what is called the **dry bulb temperature**.

A comparison of wet and dry bulb readings allows the relative humidity to be determined from a psychrometric chart. The wet bulb temperature is always lower than the dry bulb value except when the air is already saturated with water - 100% relative humidity. This is when the wet and dry bulb temperatures are the same. The air will no longer accept water and the lack of evaporation does not allow the wetted bulb to reject heat into the air by evaporation. This situation would be

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similar to operating a cooling tower at 100% relative humidity. The only rejected heat is that which is responsible for increasing the air temperature.
A single wet bulb reading will allow a prediction of cooling tower performance at that unique condition but the wet bulb changes throughout the day and year. The design wet bulb is typically determined by reviewing a chart that has been prepared by taking numerous readings in a particular area over several years and determining the maximum wet bulb readings. The wet bulb can be thought of as the heat sink temperature to a cooling tower. The lower the wet bulb, the drier the air, the more moisture it will accept and the more heat a given cooling tower is capable of rejecting.

When sizing a cooling tower, then, the highest anticipated wet bulb should be used. During the rest of the time, the cooling tower is oversized for the duty. The exiting water temperature will simply be less than design which is typically desirable. A wet bulb chart is arranged to show the frequency of occurrence. At the Charles DeGaul Airport, for example: 65 degree wet bulb is exceeded 2% of summertime hours, 68 degree wet bulb is exceeded .5% of summertime hours, and 70 degree wet bulb is exceeded .2% of summertime hours. Generally, the designer would select 70 degrees as the design wet bulb for a situation like this but some installations aren't critical allowing the use of a reduced design values and smaller cooling towers. Other installations may work only in the winter or at night when the wet bulb temperature is low. The designer must select the design wet bulb for the project. When in doubt, select the highest anticipated wet bulb temperature to insure satisfactory year around operation. The designer should only consider towers with independently certified capacities. The Cooling Tower Institute (CTI) lists towers that subscribe to their test standard STD-201. Alternately, the designer should specify a field test by an accredited independent test agency in accordance with CTI Acceptance Test Code ATC-105 or ASME PCT-23. Manufacturer's catalogs have cooling tower selection charts with easy to follow instructions that begin with the calculation of two values:

$$\text{Range} = \text{Inlet temperature} - \text{Outlet temperature}$$
$$\text{Approach} = \text{Outlet temperature} - \text{Wet Bulb Temperature}$$

These values coupled with the flow rate and wet bulb temperature allow the selection of a cooling tower. Those new to cooling towers should make several selections at different wet bulb temperatures to test how wet bulb relates to cooling tower size. It becomes clear that the tower size increases as the wet bulb rises and that the size increase becomes dramatic as the approach is in the 'less than ten degrees' area. This exercise demonstrates how to oversize a cooling tower... just use an inflated design wet bulb temperature. This is better than artificially inflating the flow rate and possibly over sizing the spray nozzles. Increasingly, manufacturers offer software to make selections easier.

Calculating the heat transfer and water evaporation rates are illustrated by the following example. A cooling tower cools 900 gpm of water from 95 to 85 °F. The problem is to determine what the heat rejection is, and also what is the evaporation rate. The heat rejection is calculated as follows:

Heat Rejection = 900 gal/min × 10 °F × 8.33 lb/gal x 60 min/hr × 1Btu/lb- °F

= 900 x 10 x 500 = 4,500,000 Btu/hr, or

Heat Rejection (Btu/hr) = Flow (gpm) × Range (°F) × 500

Evaporation Rate = Heat Rejection ÷ 1000 Btu/lb

= Flow (gpm) × Range (°F) × 500 ÷ 1000Btu/lb

× gal/8.33 lb × hr/60 min, or

Evaporation Rate (Btu/hr) = Flow (gpm) × Range (°F) ÷ 1000

A common misconception is that the cooling tower dictates the rate of heat transfer. A cooling tower simply gives up the heat it is supplied with. If the cooling tower is large, it may accomplish the job by cooling water from 90 to 80 °F. If it is small, it might cool the water in the same process from 100 to 90 °F. In either case, the heat transfer and evaporation rates are the same.

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The size of the cooling tower, the flow rate and the wet bulb temperature determine the inlet and outlet water temperatures- but not the difference between them. Increased cooling tower performance can be achieved by adding surface area or by boosting the cfm.

The former is considerably more expensive than the latter inasmuch as a cfm increase can be as simple as employing a more powerful fan motor allowing increased fan speed. Cooling towers must be evaluated on a life cycle cost basis. Spending a little more for a tower that uses less horsepower or lasts longer is almost always the wisest decision.

The most common use for cooling towers is in air conditioning as the heat rejecter in a mechanical refrigeration system. The expression 'ton' is derived from this application. Its origin goes back to the earlier days when theaters, concert halls and the like were cooled with ice. Typically, this ice was harvested from lakes and stored for summer use. At its eventual destination, it would be placed in bunkers where circulated air would melt the ice and cool the air. It is hardly imaginable that such practice was widespread in many parts of the world during the early twentieth century; nonetheless, modern HVAC (Heating-Ventilation-Air-Conditioning) technology stems from this practice, and the use of early terminology has persisted through modern times.

With the introduction of mechanical refrigeration, the term 'ton' was retained. The owner could now buy a system capable of providing the equivalent capacity of however many tons was needed. Since one pound of ice absorbs 144 Btu when melting, one ton of ice melting over a period of 24 hours has a heat transfer rate equivalent to the following:

$$\text{Heat Transfer Rate} = 2{,}000 \text{ lb} \times 144 \text{ Btu/lb} \times 1/24\text{hrs} = \underline{12{,}000 \text{ Btu/hr}}$$

Thus, when the occupants of the room experience heat removal at the rate of 12,000 Btu/hr, they are enjoying one ton of cooling. The mechanical refrigeration system utilizes a compressor that adds its heat energy- basically the motor horsepower- to the refrigerant. The cooling tower must reject not only the 12,000 Btu/hr/ton from the space but also the heat of compression as well. This added load typically amounts to about 3,000 Btu/hr/ton for air conditioning systems. So, while the occupants are enjoying 12,000 Btu/hr/ton heat removal, the cooling tower is rejecting heat at a rate close to 15,000 Btu/hr/ton. This gives rise to the term 'Cooling Tower Ton' which is defined as 15,000 Btu/hr. This definition is only valid for typical air conditioning conditions. Another very common term is 'nominal capacity'. It also has its roots in air conditioning and involves the following assumptions: The cooling tower will circulate water at the rate of 3 gpm/ton. The water enters the tower at 95 °F and exits at 85 °F. The design wet bulb temperature is 78 °F. Evaluating cooling tower capacity at the nominal conditions is an easy way to determine which tower is larger (the term'bigger' is often used) when comparisons are made. Someone employing mechanical refrigeration at 95/85 °F condenser water conditions in a city that actually has a 78 °F wet bulb temperature can use the nominal ratings as an accurate prediction for the cooling tower performance. A reduced wet bulb temperature allows the use of a smaller cooling tower. The converse is true - at wet bulb temperatures above the nominal 78 °F value... the nominal rating of the cooling tower on a 300 ton project at 80 degree wet bulb will be more than 300 tons (actually, about 360 tons).

Absorption refrigeration is another method of making chilled water for air conditioning. Instead of a compressor, the design utilizes heat energy to increase the pressure of a refrigerant. An absorption system by its nature requires a cooling tower that will remove about 50% more heat than a mechanical refrigeration system of the same capacity. The cooling tower flow rate, range, or a combination of the two must be increased so that their product is about 50% more than that for a mechanical refrigeration system.

Altitude has an effect on cooling tower performance but in a unique way. Air handlers, air cooled condensers and the like are typically made to operate at higher speeds (or, with a steeper fan pitch) as altitude increases in order to maintain the

same mass flow. As an example, air at 5,000 ft. is approximately 17% less dense than at sea level and the fan speed increases by the same amount. A cooling tower designed for operation at sea level will work just fine at 5,000 ft elevation without modification. This is because air at reduced atmospheric pressure will accept increased amounts or water. The increased ability for the air to accept more water offsets the reduced air mass resulting in a small net gain in capacity at altitude. This is why manufacturers do not make altitude corrections with their small, package towers.

DESIGN CONFIGURATIONS

Cooling towers are classified according to the method by which air is introduced to the tower. The principal types are atmospheric spray, natural-draft, mechanical-draft, deck-filled, spray-filled, coil shed and hyperbolic towers. Most industrial cooling tower installations are field-erected units designed for specific thermal characteristics. In an atmospheric spray tower the air movement is dependent on atmospheric conditions and the aspirating effect of the spray nozzles. Natural-draft cooling tower operation depends on a chimney or stack to induce air movement. Mechanical-draft cooling towers utilize fans to move ambient air through the tower. Deck-filled towers contain tiers of splash bars or decks to assist in the breakup of water drops to increase the total water surface and subsequently the evaporation rate. Spray-filled towers depend only on spray nozzles for water breakup. Coil shed towers are comprised of a combination structure of a cooling tower installed on top of a substructure that contains atmospheric section coils. Hyperbolic natural-draft cooling towers are typically large-capacity systems. There is also a separate class of towers based on a dry cooling principle.

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Cooling towers are broadly classified on the basis of the type of draft: natural draft (natural convection), mechanical draft (forced convection) and mechanical and natural. Further distinction is made based on (1) the type of flow; i.e. - crossflow, counterflow, cocurrent flow; (2) the type of heat dissipation-wet (evaporative cooling), dry, wet-dry; and (3) the type of application-industrial or power plant. Each of the major types of cooling towers has a distinct configuration. The major designs are summarized in Figures 1 through 8 and a brief description of each follows.

In an atmospheric spray towers, the air movement depends on atmospheric conditions and the aspirating effects of the spray nozzles (refer to Figure 1). In comparisons, a natural-draft cooling towers, as illustrated in Figure 2, depend on a chimney or stack to induce air movement through the tower. Both design configurations are very common in the power industry throughout the world.

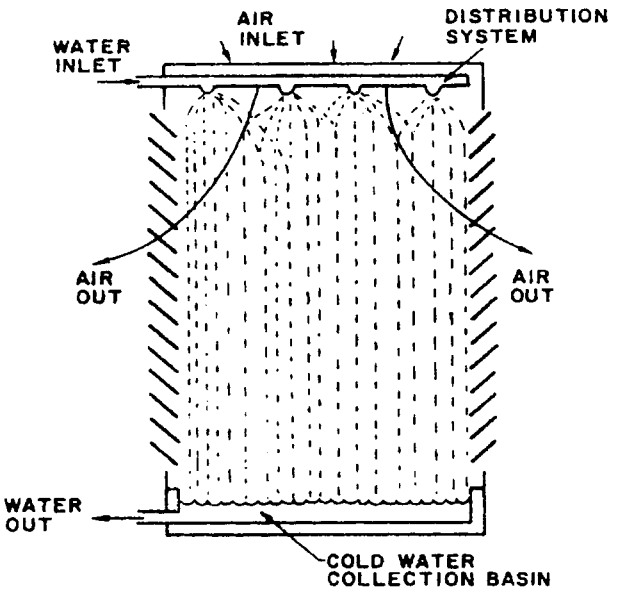

Figure 1. *Atmospheric spray tower.*

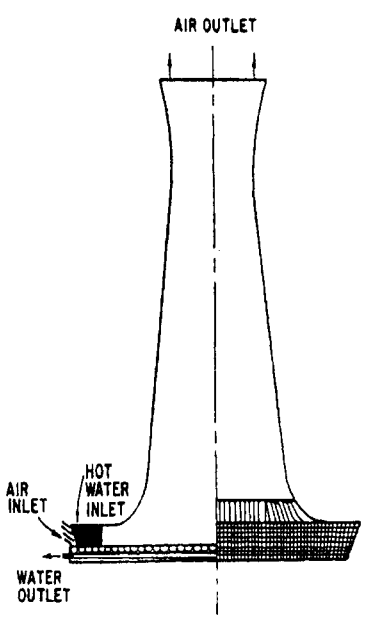

Figure 2. *Hyperbolic cooling tower.*

Mechanical-draft cooling towers use a fan to move ambient air through the tower. They can be subdivided into two different types-indirect or forced air towers, depending on whether the air is pulled or forced through the tower. At the same time they can be grouped in crossflow or counterflow, depending on the relative movement of air and water. In the crossflow type, air generally travels horizontally across the failing water, while in counterflow, it travels vertically upward through the falling water. Figures 3 and 4 provide some examples.

Another type of crossflow cooling tower is the wet-dry tower, which consists of a normal crossflow tower over which a few air coils are placed. The hot water is first cooled by an air cooled heat exchanger and then drops to the wet cooling tower where more cooling is obtained by the evaporative mechanism. Figures 5 and 6 provide examples. In contrast, deck-filled towers contain tiers of splash bars or decks to aid in the breakup of water drops to increase the total water surface and, subsequently, the evaporation rate.

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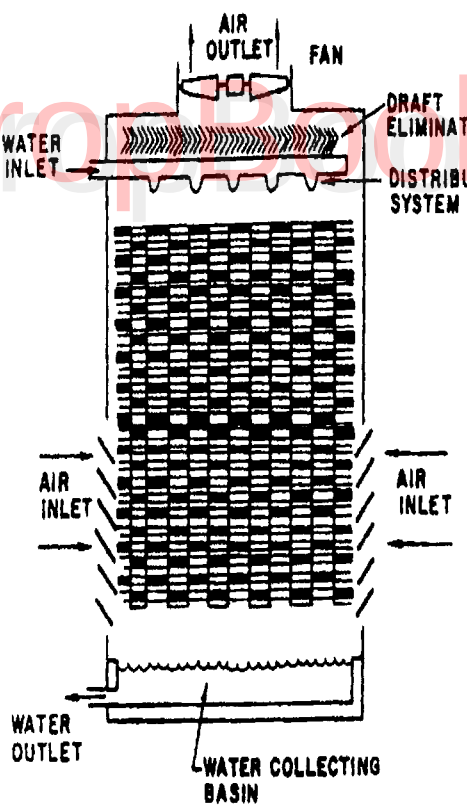

Figure 3. *Counterflow cooling tower.*

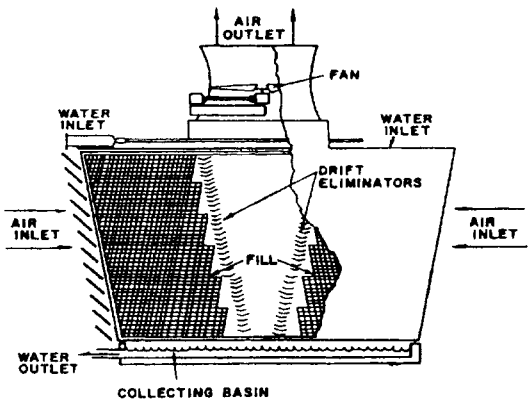

Figure 4. *Crossflow cooling tower.*

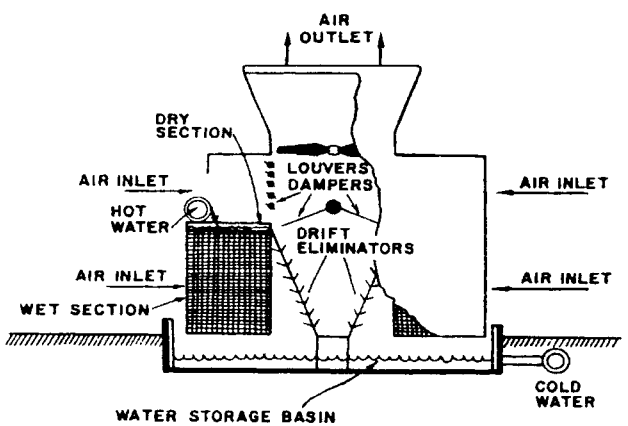

Figure 5. *Single structure type wet-dry cooling tower.*

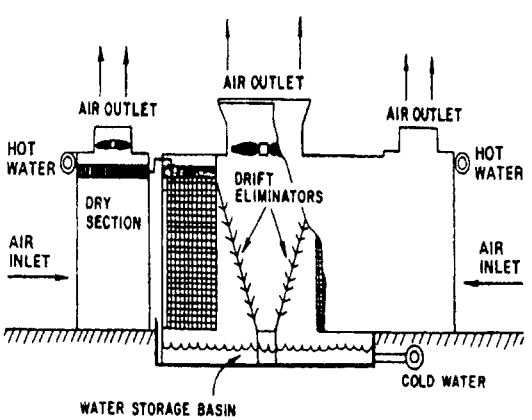

Figure 6. *Wet-dry cooling tower.*

Other common design configurations include:

- Spray-Filled Towers - These depend only on spray nozzles for water breakup.
- Coil Shed Towers - These are composed of a combination structure of a cooling tower installed on top of a substructure that contains atmospheric section coils (refer to Figure 7).
- Hyperbolic Towers - Natural-draft cooling towers with a hyperbolic configuration are usually constructed of concrete, have a large dimension and, consequently, large capacities. They are generally used in power plants. Figure 2 also illustrates this design.

Natural-draft cooling towers evolved from spray ponds. The simplest and earliest design consisted of a small water spray pond surrounded with walls having inwardly sloping louvers. Figure 8 shows a cross section of a simple cooling tower formed by enclosing a spray pond with louvered walls. In this type of design, the air changes direction in passing through the louvers as it leaves, and the suspended water droplets impinge on the louver slats, deposit outward and drain back into the tower basin. This design is improved by utilizing low-pressure water sprays at the top, positioning horizontal decks in the tower, and arranging the walls so that the air enters horizontally and is discharged vertically. The vertical upward movement of air tends to slow the downward velocity of the water droplets, thus increasing the effective surface area for any given water loading. The fill material is usually staggered so that water droplets fall through a distance of only a few feet before striking another surface. Typical grids are made of redwood strips railed to 1 x 2 inch stringers.

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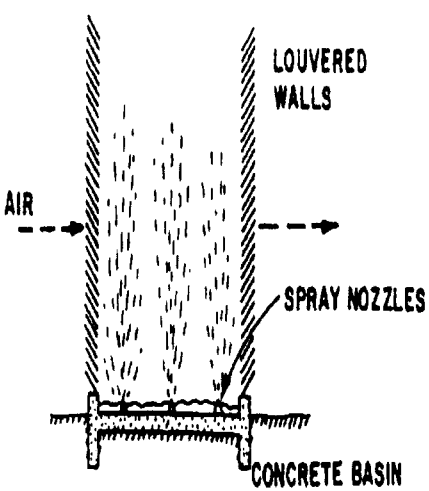

Figure 7. *Simple configuration.*

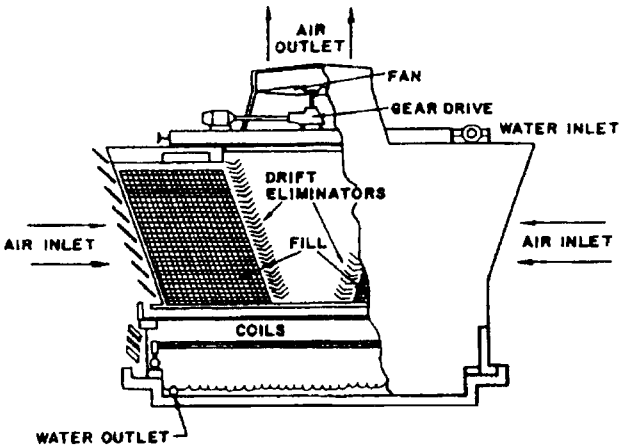

Figure 8. *Coil shed cooling tower.*

The increasing temperature of the upwardly moving air stream induces circulation by thermal convection. This is a favorable condition if the wind velocity is very low. Towers are generally placed in aside-by-side arrangement (i.e., in a row) at right angles to the direction of prevailing winds. Hyperbolic towers are an important class of cooling towers. Early designs consisted of a cylindrical configuration. Later designs used a pair of truncated cones. Today's design consists of a hyperbolic shape. The advantages of this configuration include (1) superior strength, (2) they are a close match to the natural flow of air through the tower shell, and (3) the need for fewer materials of construction than for earlier shapes (less total volume is required because the hyperbolic shape of the shell provides good strength, so a thinner shell thickness can be used in comparison to older designs). The performance of a natural-draft tower is characterized in terms of a duty coefficient (C_D), which defines the overall capabilities of a tower under all operating conditions. The draft is due to the difference between the density of the air leaving the tower and that entering the tower, and to the aerodynamic lift of the wind passing over the top of the tower. Increases in loading, cooling range and humidity all lend to improve cooling tower performance. As already noted, the two basic types of hyperbolic towers are counterflow and crossflow, as shown in Figure 9. Of the two operations, counterflow (or countercurrent) provides the more efficient heat transfer mechanism because the coolest water contacts the coolest air initially. In the crossflow operation, air flow is normal to the water movement and fill is needed to transfer a given quantity of heat. These designs have fill in a ring outside the tower. This produces a lower water pumping head than in the counterflow. Fill inside a counterflow must be spread over a much larger area. Thus, crossfiow units have shallower depths and vertical water risers have shorter pressure drops through the fill (much less than in a counterflow tower).

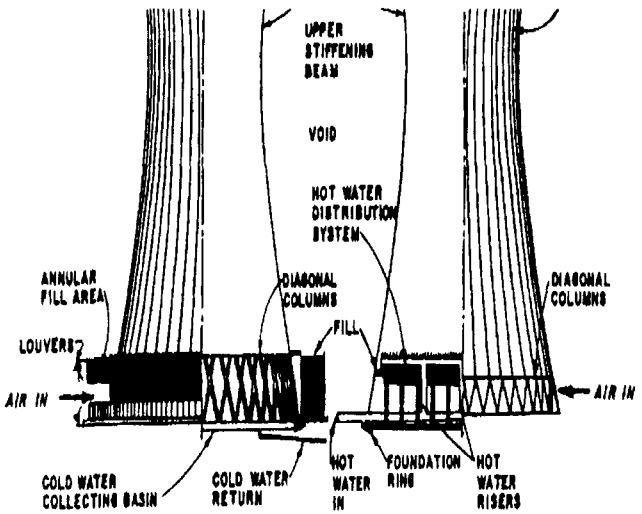

Figure 9. *Compares cross flow and counterflow towers.*

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COMPONENTS AND MATERIALS OF CONSTRUCTION

Redwood was an early construction material for cooling towers because of its natural tendency to inhibit decay. As supplies of good quality redwood diminished, douglas fir was pressed into service. Its superior strength allowed for the use of fewer structural members but it deteriorated more easily. Treatments were developed to kill the decay causing micro organisms that depleted the wood. The lumber was essentially dipped preserving chemicals which were toxic in nature. The process was enhanced by incising and pressure treatment. Incising is a procedure where small longitudinal incisions are made into the wood to encourage the acceptance of chemical treatment. Pressure treating has the wood placed in a vessel that is first evacuated to remove moisture from the wood followed by the introduction of chemicals under pressure. Simultaneous with the greater reliance on fir was the increased use of factory assembled steel cooling towers in ever increasing sizes. Galvanized steel performed well with the water treatment chemicals of the day, which were predominantly chromates and acid. Acid was used to lower the pH allowing higher cycles of concentration of the dissolved solids while chromates quelled the rampant corrosion that would otherwise occur. Asbestos cement board casing side panels were popular on field erected cooling towers. Asbestos was also found in the wet deck and sealing compounds of many towers.

Increasing environmental concerns in the 1970's brought an end to the use of chromates except for a few very large facilities where they were removed from the discharge water at site treatment facilities. Without chromates, the low pH water was very corrosive and many cooling towers and piping systems were ruined in short time. Substitutes have never lived up to chromates for effectiveness and cost. Wood towers didn't escape intense environmental scrutiny. The potential hazards of wood treatment chemicals became more apparent causing revised formulations and tighter controls both leading to increased costs. Asbestos also came into disfavor and was quickly phased out of cooling towers as the manufacturers became more aware of the potential health and financial liabilities. Type 304 stainless steel (SST) became more popular as the corrosion potential increased. Manufacturers simply substituted stainless steel for galvanized steel components. Due to cost constraints, just the cold water basin was typically upgraded to SST. There were some unfortunate occasions where galvanized and stainless steels were fastened together below the water line causing rapid deterioration of the galvanized steel at the joint from galvanic corrosion. Anyone considering mixing these materials must pay attention to the surrounding materials, particularly the fasteners. Such joints should never occur below the overflow level of the cooling tower.

Specifiers will sometimes call for type 316 SST. This is generally acceptable for nuts, bolts, and some small sub assemblies but it is largely incompatible with the tooling used by the manufacturers. It is also difficult to form. For these reasons, it is largely unavailable. The galvanized steel cooling tower has remained the factory assembled standard to this day. The thickness of the steel has steadily declined with more economical designs but the thickness of the zinc layer has steadily increased to a current standard of G235. (or, 2.35oz. of zinc per sq. ft.) from a 1970's standard of G90 (.90 oz/sq.ft.). This thickening of the sacrificial zinc layer has a very beneficial effect on cooling tower life. Various enhancements to the galvanized steel in the form of barriers have been employed by some manufacturers. Their suitability largely depends on the local water quality.

Concrete can be an excellent construction material for basins- even side walls, fan decks, discharge stacks, and mechanical support beams. Its use beyond basins, however is not typically justified for commercial applications. Extensive concrete construction is used for architectural reasons- where the tower is disguised to look like or blend in with a building- or, the cooling tower is designed as a structure with a life expectancy equal to the facility it serves such as a hospital or university.

Pultruded fiberglass is increasingly replacing steel in structural applications. These are composites with precisely located glass fibers that make the parts very strong. The process allows the addition of surface treatments that limit ultra violet degradation- an important requirement for cooling tower duty.

The wet deck or surface, is the heart of most cooling towers. Generally, it takes the form of PVC (polyvinyl chloride) plastic film type surface. Water is made to spread out on this surface maximizing it's contact area with air to encourage evaporation. It consists of individual vacuum formed sheets with proprietary patterns of ridges,

bumps and wrinkles. When arranged vertically (side-by-side), the individual sheets space themselves apart leaving passageways for water and air. The sheets can simply press against each other or be glued together. Edges can be folded for increased strength. A block of glued together film type fill can be placed on a table top and observed. If a marble were dropped through the fill it would follow one of the channels that are formed between the adjacent fill sheets. These channels- or flutes- are typically at an angle to vertical to increase the residence time of the water as it falls through the wet deck. The goal of the film type wet deck designer is to maximize air/water contact while minimizing air flow pressure drop. Reduced flute sizes increase thermal capacity limited by air flow pressure drop and clogging. Typical clean water applications such as air conditioning can have marble sized flutes. Dirty water applications like steel mills, on the other hand, may require golf ball or base ball size flutes.

Crossflow and counterflow wet decks are designed differently. Tile fill is a bullet proof approach to wet deck. It is suitable for clean to moderately dirty water and has extraordinary longevity. Heat transfer efficiency is less than that for film type wet deck, however, requiring more volume or more fan horsepower for equivalent capacity. Splash bars are another method tailored to extremely dirty water applications. Instead of spreading the water into a thin film, the approach is to have the water splash into droplets as it cascades through the tower splashing off successive splash bars. Clearly, the total surface area of all the water droplets is far less and the thermal capacity is diminished versus film type fill. On the positive side though, considerable debris can be tolerated and cleaning is relatively easy. The ultimate in dirty water towers is the 'spray fill' design. Here, there is no fill at all. Water simply sprays into the empty plenum area of a tower. This design is limited to counterflow type towers.

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Eliminators are used to remove water droplets from cooling tower discharge air by imparting several rapid directional changes. The heavier water particles collide against the eliminator and drain back into the tower. Superior eliminator designs limit escaping water droplets of the recirculated flow rate while imparting minimal pressure drop to the airstream.

The Spray tree is used to distribute water over the wet deck in counterflow cooling towers. It can consist of a single header fitted with spray nozzles or, it can utilize spray branches with nozzles for wider coverage. Spray nozzle designers seek minimal pressure requirements and uniform coverage over wide flow ranges.

Hot water basins are used to distribute water in crossflow towers. Here, water is pumped to an open pan over the wet deck fill. The bottom of the pan has holes through which water is distributed. Manufacturers will fit specially shaped plastic drip orifices into the holes to give the water an umbrella shape for more uniform distribution. Different size orifices are used for different flow rates. Ideally, the basin will be almost full at maximum flow. This way, sufficient depth is retained for good water distribution as turn down occurs. The turn down ratio can be extended by the addition of hot water basin weirs- a pattern of baffles perhaps 2

inches tall fastened to the basin floor- that insure good water distribution by selected nozzles at reduced flow. As full flow is restored, the water overflows the weirs to again engage all available orifices. Cold Water Basins collect cooled water at the bottom of the tower. They are an integral part of factory assembled designs and are built in place- typically of concrete- for field erected towers.

A make-up valve replaces water that exits via evaporation and bleed with fresh water. It operates somewhat like the valve found in a conventional toilet tank but is larger and more heavy duty. Like toilet tank floats, they can function mechanically or hydraulically. Cold Water Basin Heaters address freeze-up for cooling towers in cold climates. They are electric immersion heaters installed below the water level that add sufficient heat to prevent freeze damage. A 1,000 ton cooling tower may use something like 30 KW. This is equivalent to 30 KW x 3,415 Btu/hr/KW x 1/ (12,000 Btu/ hr/ ton) = 8.5 tons. Some designers expect immersion heaters to heat the system water as an aid to cold system start up but sump heaters are clearly inadequate for this task. Water cascading through a 1,000 ton tower- even with the fan off- will easily reject more heat than a small sump heater can possibly add. Sump heaters should be interlocked with the system pump and only operate when the pump is idle. Properly configured controls will include a thermostat typically set to energize the heater when the sump water temperature is below 40 °F. Steam or Hot Water Coils can also be employed in lieu of electric heaters.

It is never appropriate to add any type of anti-freeze solution to an open cooling tower. Closed (fluid cooler) systems, however, can be protected from freeze-up by the addition of ethylene glycol or other fluids. Fluid cooler casing sections can also be insulated to reduce heat loss thereby protecting the coil from freeze-up. Counterflow, blowthrough towers tend to be more popular as the freeze potential increases. Crossflow towers tend to freeze water on their air inlet louvers under extreme conditions. Fans (propeller type) can be arranged to reverse direction on such towers to melt ice. This process should never be automated. Instead, the operator should weigh the situation and reverse the fan only as long as required. The designer must select components suitable for reverse rotation. Fan discharge dampers are a capacity control accessory item for centrifugal fan cooling towers. They fit in the fan scroll. In the open position, they are much like a thin piece of sheet metal in a moving airstream oriented parallel to airflow. The airstream doesn't know its there. As the dampers close- the sheet metal becomes less parallel to airflow- turbulence disrupts the air stream. Airfoil dampers essentially ruin fan housing efficiency to achieve a reduction in airflow. Dampers can set and locked when a manual locking quadrant is specified but it is more common to use electric or pneumatic actuators that close the dampers as the exiting water temperature becomes too low. While reducing airflow is the correct method of reducing capacity, dampers are not the best approach. They offer the poorest energy savings and the actuating mechanisms tend to fail long before the average cooling tower life span.

System designers often think dampers block airflow and are suitable to prevent back drafts in idle towers. This is not the case. Airfoil dampers simply hamper fan housing efficiency- they do not block airflow. Air Inlet Screens are always part of blow through, counterflow towers to protect people from rotating equipment . Some designs can be a hazard when accessible from the underside and require the specifier to call out additional screening. They can be a worthwhile accessory when there are nearby trees even when not required for safety reasons. Air inlet screens should be eliminated on towers utilizing inlet ductwork. Inlet ductwork may also make it necessary to block extraneous air entry such as from the underside when towers are elevated.

A vibration cutout is a control device used to shut down the fan motor when excess vibration is sensed. They can be used on any tower with a fan motor either by choice or by code although they are only practical for towers employing large propeller fans. Typically, centrifugal fans do not fail in a catastrophic mode; Similarly, small prop fans don't cause enough damage to require such devices. Ladders and handrails are also necessary components for large field erected cooling towers and make sense on some factory assembled designs. Often, just a ladder makes more sense. Or, nothing at all on small towers. Internal Ladders, Walkways, Platforms, etc. should be evaluated on a job by job basis.

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USE OF FANS, MOTORS, AND DRIVES

Cooling tower fan motors must be properly selected for long life and trouble free operation. In addition, it is important to match the motor's enclosure to the application. Motors in draw through applications that are mounted directly in the air stream, for example, experience tough duty and must be of the totally enclosed type. Motors are categorized as follows:

- TEAO - Totally Enclosed Air Over where the motor has large cooling fins and depends on the cooling tower air stream for air movement. The motor shaft protrudes from the enclosure at one place only.
- TEFC - Totally Enclosed Fan Cooled where the shaft extends from both ends with a cooling fan attached to one with a shroud that directs air over strategically located cooling fins. This motor is more commonly used when the motor is outside the air stream as when driving a gear box with a drive shaft or on blow-through towers.
- ODP - Open Drip Proof where there are openings to the windings through the enclosure and a cooling fan inside that causes air to flow through the motor. ODP motors should never be installed in a discharge air stream. They can be placed in the inlet air when located in such a way as to be protected from

splash out and rain. TEFC motors should be the minimum standard. TEFC motors in the discharge air stream with the shaft pointing down- as with most belted applications- have an increased possibility of moisture entering the motor around the cooling fan.

In addition, the small motor fan attempts to blow air down over the motor and is no match for the much bigger cooling tower fan drawing air up and around the motor. The small fan in this case is useless- even counterproductive. This application needs a TEAO motor. Cooling tower manufacturers sometimes purchase special motors tailored for their application... special grease, seals, slingers, weep hole locations, epoxy coatings, etc.. Such features make an OEM replacement more desirable than an 'off the shelf' replacement.

One advantage of having the motor in the air stream is the generous cooling that it receives. A motor rated at, say, 60hp in a 'normal' application may be capable of providing a continuous 72hp without any difficulty because of this cooling. This is why it is important to size conductors, fuses, starters, etc. based on actual motor amps (which the manufacturer should provide in his quotation) and not from standard application charts.

Large towers often have the motor mounted horizontally connected to a right angle gear drive. The motor can be closely coupled- in the air stream- or connected with a drive shaft with the motor outside the air stream. Maintenance personnel typically prefer the external TEFC motor- when available- for its easier access. Cooling tower fans- like all fans- operate in accordance with the fan laws one of which states that the horsepower required to drive a fan increases to the cube of fan speed.

$$HP_2 = HP_1 (RPM_2 / RPM_1)^3$$

As an example, consider the speed of a fan that is increased by 10%. Let's determined the revised horsepower requirement.

$$HP_2 = HP_1 (RPM_2 / RPM_1)^3 = HP_1 (1.1 / 1.0)^3 = 1.33HP_1$$

The horsepower increased 33% while the speed went up by only 10%. Similarly, slowing the fan by a small amount causes a marked reduction in the horsepower and amperage draw. It is apparent that speeding up a cooling tower to increase its airflow and capacity must be carefully approached. Aside from possibly exceeding the maximum safe speed for the fans or causing the increased airflow to 'fling' water past the eliminators - a phenomenon referred to as 'spitting'- the required horsepower climbs very quickly requiring oversized motors with attendant energy costs for only modest capacity gains. This fan law demonstrates another important fact, namely, that fan motors have small torque requirements at start up and don't begin to really work until they get near top speed. Fan motors, therefore, do not need special starting schemes such as part winding or Y-start, Delta-Run. Fan

motors are simply started across the line. Occasionally, motors show optional starting features on their name plates causing operators to seek out the special starters depicted for no reason. This is simply a case where the motor manufacturer stocks motors with a wide application range. Cooling tower fan motors are often used to modulate air flow through a cooling tower to kill off excess capacity during periods of low load, or especially during winter operation. This capacity reduction can be achieved with fan cycling, multi-speed motors, extra motors, or variable speed drives. Fan cycling (turning off fan motors) works well when a tower has numerous fan motors. If there are four fan motors, for example, turning one motor off reduces the capacity by about 1/4. This is an easy capacity control method but doesn't work well when close temperature control is required resulting in frequent motor starts. As a rule of thumb, six starts per hour should be considered maximum. Excessive starting causes heat build up and insulation failure. Prolonged "pump on, fan off" operation is not good and should be avoided. Two speed fan motors are available as either single or dual winding. The single winding motor has its entire winding active at low or high speed. The winding is simply reconfigured by the starter as either 8-pole or 4-pole (900rpm or 1800rpm). Low speed is always half of full speed. These motors are wound for a specific voltage- most often 460v. The two winding motor has two separate and distinct windings- one for low speed and the other for high speed. It is possible for one winding to fail leaving the other intact but such occurrences are rare. Typically, a faulty winding takes out its neighbor. And, since the motor has to be removed for repair anyway, there is no real standby advantage to such a motor. One advantage of a two winding motor is that the speed ratio is not necessarily 2:1. Common speeds are 1800/900 and 1800/1200. These motors are also wound for a specific voltage. In general, single winding motors cost less but their starters cost more. Conversely, two winding motors cost more and their starters are less expensive. In the end, there is little cost difference. Single winding motors are more likely to be stocked and are far more popular. Either should be specified as variable torque.

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Pony motors are additional, small motors connected to the same fan shaft. They are typically about 1/4 the size of the full size motor. Pony motors do not lend themselves to gear box applications and are, therefore, primarily applied to belt drive applications. The big motor operates when full capacity is required and the small motor simply free wheels. At reduced capacity, the small motor operates and the big motor spins freely.

The appropriate drive ratio is selected for each motor so that it is fully loaded when in operation- a distinct advantage over two speed, variable torque motors where the available horsepower is proportional to the square of fan speed while the required fan horsepower varies as to the cube of fan speed (from the fan law). As an example, a fan motor that can produce 40 hp at high speed can produce 10 hp at low speed while a fan that requires 40 hp at high speed only requires 5 hp at half speed. As a result, 1800/900 rpm motors are always 100 % oversized at low speed. And, since the motor usually operates at low speed most of the time, the

inefficiency of the lightly loaded motor is noticeable. The pony motor also has the advantage of allowing one motor to be removed for servicing while the other remains on line. Plus, such simple, single speed motors are readily available in the event of a breakdown. Electrically, the pony motor arrangement is equivalent to a two winding motor. The two windings physically reside in separate motors.

Questions arise as to the 'idle' motor acting as a generator when rotated by the active motor. Aside from other differences between motors and generators, the fact that there is no excitation current means that there can be no output. The only losses seen by induction motors are from windage and belt flexure and are so small as to be virtually undetectable.

Single phase motors do exhibit operational problems in pony motor applications. The capacitors in these motors store energy and the motors resist external attempts to rotate them. Single phase motors can be made to work on pony motor applications by splicing into the capacitor circuit and connecting it in series with an open auxiliary switch in the active motor starter.

Examination of the drive ratios shows that if both motors have 1800rpm synchronous speeds that the pony motor is made to operate near 3600rpm when the big motor is operating. This is typically not a problem as to rotor balance or bearing duty because manufacturers make 3600 rpm versions of these same motors. Nevertheless, the duty should be checked and if this is a problem, the small motor can be changed to a 900 rpm model and the synchronous speeds of each motor will not be exceeded.

Two speed motors and pony motor arrangements both require a time delay that prevents low speed or pony motor operation until approximately 15 seconds after high speed operation. This insures that the low speed winding is not energized while the motor is rotating faster than its synchronous speed. For example, an 1800 rpm pony motor rotates at about 3600 rpm when the 'big' motor is operating. If the 'big' motor is switched off and the pony motor switched on immediately, it would attempt to operate at its 1800 rpm synchronous speed but would already be rotating closer to 3600rpm. The conflict will cause a deceleration of the rotating components so severe as to possibly cause damage. The 15 second time delay insures the motor is rotating under its synchronous speed when energized. It will then gently accelerate to full speed. Fifteen seconds is a starting point; The time delay can be reset to a lower value appropriate for each project.

Variable speed drives are the ultimate in capacity control but introduce a level of sophistication that may not be required. Projected fan motor energy savings make their use attractive, however there are other considerations. Often the excess energy consumed by a single speed motor is more than offset by increased system efficiency from the reduced water temperature provided by the cooling tower. When used, VFD's and two speed motors should generally be set to provide the coldest temperature that the system will tolerate before reducing motor speed. VFD's can, however, be very helpful in noise sensitive applications. Soft starting and gradual speed changes make cooling tower noise less noticeable to critical

neighbors. Towers that operate at extremely light loads can also benefit from VFD's. The VFD will keep a motor running with a positive air flow through the tower. This avoids excessive 'motor off' operation and attendant water 'splash out' problems. Note also that extremely low speeds can defeat the 'sling' lubrication employed in gear boxes requiring the drive to be programmed to avoid low speeds. Alternately, an electric oil pump can be applied to insure adequate lubrication at all speeds.

A common problem encountered is 'windmilling'. This is when the cooling tower is baffled in such a way as to allow air to pass in a reverse direction past a supposedly idle fan when its neighbor is in operation. In this situation, the 'idle' fan rotates backwards. This causes numerous problems not the least of which is tremendous stress on the drive components when they attempt to start while rotating in a reverse direction. Economic design is often responsible for the omission of baffles. The designer should not assume that just because there may be multiple motors that they can all be operated individually. Anti windmilling devices- essentially one way clutches- are available to some applications but are not necessarily the best solution. It is better to baffle the tower properly so that each fan can operate independently. When replacing fan motors, be sure to match all the nameplate characteristics such as hp, rpm, voltage, phase, frame size, enclosure, service factor, insulation class, group, etc.. Also check that the conduit box is located in the same place. NEMA -the National Electrical Motor Association- provides standards for motor manufacture. Motor frame size is one important NEMA standard. A 284T frame motor from one manufacturer, for example, will have the same bolt pattern, shaft elevation, diameter, key size, etc...among all manufacturers. UL -Underwriters Laboratories- examines components from the various motor manufacturers and publishes a list of UL recognized motors. This prevents UL from having to check each motor that they encounter when evaluating equipment in the lab. Most cooling towers are not UL listed. This is because the smoke and debris that would result from a motor failure is not directed into occupied spaces. UL. listing is therefore not required.

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Cooling tower fans can be directly coupled to the motor, or, connected via gear boxes or V-belts. Direct drives are limited to small diameter fans. Otherwise, tip speeds would be too great- even with 900rpm motors. Direct drives are, therefore, generally found on small cooling towers; or, medium size towers with numerous motors. V-belts are used on the full range of factory assembled towers- from less than 10 tons to over 1,000 tons. They allow the manufacturer to easily tailor the speed of fans through the judicious selection of sheave (pulley) diameters. Fixed pitch prop fans can be used. V-belt drives do not require precise alignment and work fine after the rigors of shipment and rigging. Belts and sheaves come from numerous manufacturers.

Gear boxes are used on just about all large field erected cooling towers and many factory assembled towers- all of the propeller type fan, draw-through type. Almost all are right angle gear drives with the input shaft horizontal and the output

vertically 'up'. The designer selects the ratio closest to his/her requirement based on the fan being used, noise levels and cost. Limited ratios are available and the fan blades must be 'pitched' to the proper angle to achieve the desired performance for the specific project- typically not a problem since large fans are built on site anyway.

Speed reducers should be right angled gear drive type specifically designed for cooling tower service. Gears should be spiral bevel or helical type designed in accordance with the Cooling Tower Institute and American Gear Manufacturer's Association Standards with a minimum service factor of 2.0 based on motor horsepower. The gear should be suitable for both forward and reverse operation. Oil fill and vent lines should be extended outside the fan stack and provided with an oil level sight glass to facilitate routine inspection and maintenance. Fan motors can connect directly to the gear box or through a drive shaft. Either way, motor and gear alignment are critical and mountings must be solid. Various manufacturers make flexible couplings specifically for difficult cooling tower duty. It is in the cooling tower manufacturer's best interest that these items work properly and be trouble free.

As with any mechanical equipment, cooling towers can generate objectionable noise and vibration. Large installations are typically laid out by experienced designers. They gather equipment away from noise sensitive areas, add concrete walls as barriers and employ acoustical consultants when necessary. Sound problems are generally infrequent but can be difficult to manage. Minimally designed condominiums or industrial plants that boarder residential areas are typical. Sound complaints don't always involve rotating machinery. The designer must strive to reduce changes in noise level in noise sensitive situations. Motors starting or changing speed, belts squealing and the like all draw attention to the tower. Variable speed drives or 'soft starters' can help to minimize these problems. Two identical towers side by side can exhibit the phenomenon where sound pressure levels are slightly in and out of phase causing an annoying addition of sound pulses- like that of a twin engine turbo prop commuter plane. Intentionally making the fans operate at different speeds through the programming of VFD's- or simply changing one of the sheave diameters slightly on a belt drive tower can help. Factory designed sound attenuators that attach to the tower can be helpful in difficult situations; However, they tend to be expensive and generally require more fan hp. So, it is better to first consider oversizing the tower and slowing the fan before employing attenuators. Vibration can also be annoying and difficult to solve. When towers are on floors or rooftops with people below, it is wise to employ spring type vibration isolation. The amount of static deflection is an indicator of isolation efficiency- generally 1" minimum and 2" maximum. Another consideration when using vibration isolators is to first mount the tower on a sturdy frame. Then, place the isolation underneath. This is especially critical for multi-cell towers. All cells must be mounted on a single frame before being isolated. Otherwise, the reduced water weight that occurs when valving off and draining one cell allows its springs

to extend causing tremendous stresses to flume boxes and interconnecting piping. Rubber pads offer a small measure of vibration isolation and work well when placed under a uniformly distributed load. However, some tower designs have highly concentrated load points that will cut through the pads. Worse yet, the underside of some towers can become distorted when they exhibit variable loading characteristics- much like a person attempting to sleep on an overly soft mattress. Rubber pads should be avoided unless the designer is sure they are appropriate. After taking the effort to isolate the tower, it is mandatory that the piping have flex connectors.

WATER TREATMENT SERVICES

Water treatment is a necessary adjunct to evaporative systems. Water quality must be maintained to prevent scaling, corrosion and biological fouling or attack. Scale and corrosion are generally thought to be diametrically opposed to one other. Reducing scale build up, for example, exacerbates corrosion and vise versa. The biological aspect of water treatment comes from living organisms that thrive in the recirculated water and wetted surfaces. Bacteria, slime and algae can foul heat exchanger surfaces and in some cases attack and destroy system components. Chemical treatments address biological issues separately from scale and corrosion. Scale formation has its root in the evaporation of water. Evaporated water exits the system as pure vapor leaving the solids behind. The replacement (make-up) water introduces more solids which continually increase the solids concentration in the recirculated water. Left unchecked, the system would reach a point where the water could not hold all of the solids in a dissolved state. They would begin to precipitate out of solution as scale. The necessary steps to combat scale differ by geographic location. Evaporating enough water to make the solids increase to twice their initial value is a two fold increase in solids content. Clearly, water that has few initial dissolved solids can attain a very high number of 'cycles' before the solids precipitate from solution. Conversely, water with high initial solids can only be 'cycled' a small amount before precipitation occurs. In either case, a saturation point will be reached where the cycles cannot be increased. Every particle that dissolves is offset by another particle that comes out of solution as scale. The water treatment service provider is aware of the dissolved solid content of the water in his/her area and knows how many cycles are acceptable. When in doubt, a make-up water sample can be taken to demonstrate the quality of the water entering the system. Each constituent is examined against a maximum allowable concentration. Some locations may require that the cycles be set based on keeping some other constituent- like silica- below a maximum threshold value. Recognizing the concept that the fewer the initial solids the better, some operators are tempted to use soft water as make-up theorizing that since most of the solids are removed, the cycles

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can be allowed to reach astronomical levels without scale formation. This approach is ill advised without the input of a competent water treatment expert who is capable of combating the excessive corrosivity of such water. In addition, the use of water treaters adds chemicals that allow the water to retain increased solids.

Water treatment controls are set to maintain the design value. This could be a 'continuous bleed' where a set portion of the recalculated water is intentionally wasted to the drain. The make-up float valve introduces fresh water to replace that which is evaporated and bled. The new water mixes with system water diluting the solids concentration. While initially inexpensive, the continuous bleed is rarely used and, in fact, is often illegal. Here, a valve is set to waste water to the drain at a rate necessary to maintain the design cycles at maximum evaporation. The problem is, towers typically aren't called upon to evaporate water at the maximum rate most of the time allowing the cycles to plummet, wasting water. More common is the use of a 'conductivity monitor' which operates on the principle that the conductivity of water increases in direct proportion to its solids concentration. The device is first used to measure the conductivity of the make-up water and then set to initiate a bleed cycle when the system conductivity reaches a value equal to this initial reading x cycles. Scale and corrosion inhibitors are typically injected into the system as it bleeds. Liquid chemicals are introduced by small, adjustable, plastic, positive displacement pumps that meter precise dosages.

The tendency for any system to grow biological material depends on several factors. Cooling tower design is one. Crossflow towers and counterflow towers without louvers, for example, tend to grow more algae due to the increased amounts of sunlight in contact with the system water. Water quality also comes into play. Make-up water that is reclaimed from a sewage treatment plant, for example, can be rich in nutrients. Also, some food processing operations where beer, tomato paste, milk, sugar, etc. enter the cooling system can have severe corrosion and biological problems. Another potential lies with air quality. Cooling towers located near bakeries, for example, show an increased tendency to grow biological material due to the molds and yeast. Biological concerns run the gamut from nearly zero to very substantial. Whatever the case, an appropriate solution must be developed. The traditional approach is for the operator to alternate between two liquid biocides adding them at a predetermined frequency. Two different formulations are often used to avoid an immunity being developed to just one. Unlike scale and corrosion chemicals that are metered into the system frequently, biocides are typically administered every few days to 'shock' the system. Other chemicals for biological control include chlorine, iodine, bromine and ozone. These are not rotated with other chemicals; They are fed by themselves. Ozone is also used to prevent scale. Be certain to specify Viton pump seals when using ozone; Standard seals will fail quickly.

The cooling tower has the misfortune of being a handy receptacle for the addition of water treatment chemicals which are almost always corrosive in concentrated form. The point of injection of scale and corrosion chemicals is important. They

should not be dripped into the top of a cooling tower where the air can blow them against metal components. They should be introduced into the piping where they will disperse quickly. Similarly, pelletized chlorine tablets tossed into the sump can burn holes through the basin floor.

There is also a general failure to recognize that cooling water quality can be very dynamic. Do not, for example, make the mistake of installing a new tower, placing it into operation, and ignoring the water treatment for a few days. Some closely coupled systems with small water volumes (evaporative condensers and fluid coolers lending the best examples) can be scaled in a matter of hours.

Cooling towers do not suffer from the accumulation of scale as quickly as their evaporative condenser and fluid cooler counterparts. Scale can be allowed to remain in a cooling tower if it is not so thick as to inhibit airflow. Just descale the device being cooled- typically a condenser bundle- by chemical or mechanical means. When cleaning chemically, isolate the heat exchanger and circulate the chemical solution through it with an auxiliary pump following manufacturer recommendations. Evaporative condensers and fluid coolers have been successfully descaled using a five percent (by weight) solution of inhibited sulfamic acid. It must be monitored very carefully- ask the manufacturer for recommendations.

Filters are an additional water treatment option, opten used with chemical treatment. A 100 ton cooling tower processes about 40 tons of air in an eight hour period retaining all the air borne debris in the tower water. Ideally, this debris would stay in suspension and be removed by the bleed but a good portion typically manages to settle in the tower basin and on heat exchange surfaces. Such debris tends to reduce the effectiveness of water treatment chemicals. Full flow filtration is generally limited to devices that operate with low pressure drop and those that have an ability to purge debris while in operation. Strainers that swing into the flow stream while others backwash and centrifugal separators can be used full flow. Another option is a centrifugal separator, which directs the water flow tangentially into a cylinder causing the water to rotate.

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Debris particles that are heavier than water migrate to the outer wall area and slither down into a collection bowl that is periodically blown down. Very little water is lost. Typical performance is 97.8% of solids with specific gravity exceeding 1.2 down to 45 microns. Side stream filtration is more common in cooling tower applications. The scheme uses a small dedicated pump to draw dirty water from the sump, direct it through a filtration device and sent it back to the basin.

In addition to being smaller and less expensive, side stream filtration has the advantage of allowing the return water to be routed through a perforated PVC distribution pipework or- better yet- nozzle jets that agitate the water at the basin floor placing the debris in suspension and increasing its chances of being drawn into the filter suction piping. The actual side stream filtration device can be a centrifugal separator, strainer device or sand bed filter. The sand bed filter is especially effective in removing particulate matter- even light weight biological material killed

by biocides. Filtration to ten microns is commonplace. When the sand begins to get clogged, a pressure differential switch signals a backwash cycle that lasts about three minutes. The valves reposition to reverse the flow through the sand bed lifting it and carrying off debris to the drain. Side stream filtration devices applied to open cooling towers are typically sized to pass the entire system volume at least once each hour for good water clarity. A second filter selection method can be used when the system volume isn't known- and on all evaporative condensers and fluid coolers- that relates the filter size to the amount of air ingested:

$$\text{Filter GPM} = \text{Cooling Tower CFM} \times .0004$$

Use both methods when possible and use the largest as a starting point. Then, apply a judgment factor increasing the size to accommodate blowing dust or sand. One complaint water treatment personnel have with filters is they discharge system chemicals during backwash and therefore prefer backwash be accomplished with city water. The city water backwash example is actually rare. It is far more common to employ system water for backwash. City water backwash is really only necessary when sumps have insufficient volume to keep from running out of water during the backwash cycle. From the filter supplier's side, city water pressure can be too high or too low causing the backwash cycle to be too vigorous or too lethargic. They feel more comfortable with the known pressure generated with their own pump delivering 'system' water for backwash. Filter backwash doesn't diminish the residual chemical level appreciably when system volumes are large; However, closely coupled systems with 'small basin' cooling towers can suffer severe chemical depletion from backwash. There should be no reason why controls can't be provided and set to prevent backwash until it is appropriate to bleed and to feed the appropriate chemical dosage after bleeding.

GLOSSARY OF TERMS

There are a number of terms with rather rigorous definitions within the context of cooling tower technology. It is worthwhile to learn these definitions.

Acceptance Testing: Test procedures to determine the water cooling capacity of towers. Instrumentation used and measurement procedures should be those recommended by the Cooling Tower Institute (CTI) in its "Acceptant Text Procedures. "

Air Inlet: That portion of the cooling tower structure in which air is drawn into the system.

Ambient Dry-Bulb Temperature: External outdoor temperature as indicated by a dry-bulb thermometer and expressed in degrees Fahrenheit.

Ambient Wet-Bulb Temperature: The temperature in degrees Fahrenheit to which air can be cooled, making it adiabatic to saturation by the addition of water vapor. In practical terms, the wet-bulb temperature is the temperature indicated by a thermometer, the bulb of which is kept moist by a wick and over which air is circulated.

Approach or Approach to the Wet-Bulb: The difference in temperature (°F) of the cold water leaving the tower and the wet-bulb temperature of the ambient air.

Balancing Valve: Hand or mechanically operated valve installed in each riser pipe of a multicell tower to control water flow.

Basin: The area at the bottom of the tower for collecting cold water. Crossflow towers have a hot water distribution basin at the top and, in some cases, a water basin between the top and bottom basins.

Blowdown (Purge): The continuous or intermittent wasting of small amounts of circulating water. Its purpose is to prevent an increase in the concentration of solids in the water due to evaporation, normally expressed as a percentage of the water being circulated.

Capacity: The average amount of water circulating in the cooling system at any given time, expressed in gallons per minute.

Casing: The vertical enclosing side- or endwall of a tower, exclusive of the air inlet louvers.

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Cell: A unit consisting of a distribution system, mechanical equipment and partition walls. A single tower can have several independent cells. Individual cells can be shut down, or several cells can be run on partial capacity.

Cellular Film: Asbestos fill packing that converts water droplets into a thin molecular filter for more efficient cooling. Less static pressure is encountered than with splash-bar fill, thus permitting the use of higher air velocities.

Concentration Cycles: Comparison of dissolved solids in makeup water with solids concentration in the circulating water.

Cooling Factor: The ratio of the pounds of water circulated per unit of time to the pounds of dry air cooling the water per unit of time.

Cooling Tower Institute: International organization of cooling tower engineers, manufacturers and users, dedicated to improving the professional and technical state of the art.

Counterflow: A system in which air encounters the hot water at a 180° angle. Air enters near the base of the tower and moves upwards through the fill and falling water.

Crossflow: A system in which air encounters the hot water at a 90° angle. Air enters through the entire sidewall and moves horizontally through the fill and water.

Delta Temperature (Range): Difference between entering and leaving water temperatures.

Design Conditions: Thermal parameters for which the cooling tower is purchased. They consist of a given gpm flow of water entering the tower at a specific temperature, cooling through a given range, leaving the tower at the required temperature, and having a designated approach to a stated wet-bulb temperature.

Diffusion (Redistribution) Deck: A device below the hot water distribution basin of a crossflow tower to break up the water going through the orifices before it goes through the fill.

Distribution System: Mechanical method of passing hot water over the fill uniformly. Low-pressure spray-through piping and nozzles are usually used in counterflow towers; gravity drop is normally utilized in crossflow towers.

Drift: Entrained water droplets that escape from the tower with the exhaust air expressed as a percentage of water circulated.

Drift Eliminator: Baffling that causes discharging hot air containing entraining water droplets to change direction a number of times. Droplets hit the eliminator surface and fail back into the tower.

Fill Packing: Specially designed baffling used to provide a large surface area for heat transfer. Two classes of materials are used: splash bars of wood, metal transite or plastic and film pack (cellular fill). The splash type cools the water as the droplets bounce down a series of bars in the air stream; film packing converts droplets into a thin film.

Fog: A mist formed where the ambient air cannot absorb all the plume's moisture. The intensity of the fog is a function of the heat rise of air passing through the tower and the temperature and humidity of the ambient air. Fog plumes are normally permissible since there are no droplets of water raining out of the discharge area; however, fog may cause icing of nearby roads and may restrict visibility.

Forced Draft: Air introduced at the bottom of the tower is forced to the top by a centrifugal blower.

Heat Load: Amount of heat (in Btu) dissipated in a cooling tower. It is equal to the weight of water circulated per unit of time multiplied by the cooling range.

Induced Draft: Air mover, usually an aerial fan, on top of the tower pulls air up through the fill and out the stack.

Latent Heat of Vaporization: The heat required to change a liquid into a vapor without a change in the temperature or pressure.

Louvers: Baffles used for changing the direction of air flow into the tower in a uniform, parallel manner. They are also used in preventing water droplets from

splashing out of the tower as they fail through the structure. The same principles that apply to controlling air flow with fans applies here.

Makeup: This term refers to the water required to replace the circulating water that is lost by evaporation, drift, blowdown and leakage. It is expressed as a percentage of the water circulated and normally is automatically controlled by a float valve.

Net Effective Volume: A portion of the total structural volume in which the circulating water is in intimate contact with the air flowing through the tower (expressed in cubic feet).

Performance: The measure of the tower's ability to cool water. It is usually expressed in terms of cooling a quantity of water (gpm) from a specified hot water temperature to a specified cold water temperature at a stated wet-bulb temperature.

Performance Curve: A graphic representation of the relationship of water temperature, approach, wet-bulb temperature, range, static pressure and air movement.

Plenum: An enclosed chamber in which pressure is higher than atmospheric pressure.

Plume: Visible manifestation of water vapor condensing as warm, moist air mixing with the cooler, outside air.

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Psychrometer: Instrument used to measure the wet-bulb temperature.

Pumping Head: The energy required to raise water to the distribution elevation and overcome friction losses through pipe, valves, fittings and nozzles. It is expressed in feet of liquid the pump must move and is equal to the total friction loss, static head and pressure drop through the distribution system.

Range: The numerical difference between the temperature of the hot water entering the cooling tower at the distribution system and the temperature of the colder water leaving the sump basin.

Recirculation (Recycle): Hot exhaust air forced downward and back into the cooling tower raises the wet-bulb temperature of the entering air above dry-bulb temperature, impairing tower performance. It is usually caused by design, wind or placement problems.

Sensible Heat: The heat required to change the temperature of air or water.

Static Pressure Drop: The reduction of air movement through the tower resulting from resistance of internal components such as air-intake louvers. Oil packing, water distribution system, internal supporting beams, drift eliminators and fan stack configuration.

Sump: Depressed section of the cold water collecting basin, from which cooled water returns to the heat source.

TDS: Total dissolved solids contained in solution in the cooling water system.

Water Load: Circulating rate of water over the tower, expressed in gallons per minute.

Wet-Bulb Temperature: The temperature of saturated air. The lower the wet-bulb temperature, the more exchange of heat a cooling tower can do. A tower cannot cool the water to a temperature below the wet-bulb temperature of the entering air.

Windage: The loss of water through the air-intake louvers as a result of malfunctioning of the wind check walls in the lower section of the tower.

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Chapter 3
EVAPORATING AND DRYING EQUIPMENT

INTRODUCTION

Evaporation and drying are standard unit operations. Evaporation is used in such applications as solvent recovery, product preparation, recovery of valuable by-products, in waste processing. Drying is a separate and distinct unit operation, but can be coupled with evaporative techniques depending upon the application. Drying can be used in similar applications to evaporation, but more often as a principle unit operation aimed at product finishing and waste processing. Specific examples and equipment configurations are discussed in this chapter. Often equipment designs are very unique to the application, and therefore the specifier must have a keen understanding of the nature of the operation and the process materials handled.

EVAPORATORS

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Evaporators are used extensively throughout the chemical process industries for applications ranging from concentrating black liquors for recovery furnaces in paper manufacturing, to various product drying operations in the food industry, to the manufacturing of a wide range of semi-dry and low moisture content chemical feedstock products and intermediates. Refer to Figure 1 for a simple classification system of evaporating equipment. Concentrating a solution by the evaporation of part of the solvent is a widely practiced. When the solvent is water, the function of the evaporator is to vaporize part of the water, resulting in a concentrated solution. The vapor is of no use except, perhaps, for its heat value, and it is the residual solution that is saved. This is the distinction in general terms between evaporators and a still. In the latter it is the vapor that has value, while generally, though not always, the residual liquor does not. Heating is usually accomplished with steam at low pressure. Typically saturated steam is used without superheat. Superheat does not furnish sufficient heat units to be of importance (e.g., total heat content of saturated steam at 40 lb gauge is 1169.4 Btu/lb, of which 933.3 Btu is latent heat, the portion generally utilized); while 40 lb gauge pressure steam with 100' of superheat has a total heat content of 1218.4 Btu; hence, this adds 47 Btu additional to latent heat, which is the same as for saturated steam. This heat is known as the latent heat of steam, also the heat of condensation, that vaporization is delegated to.

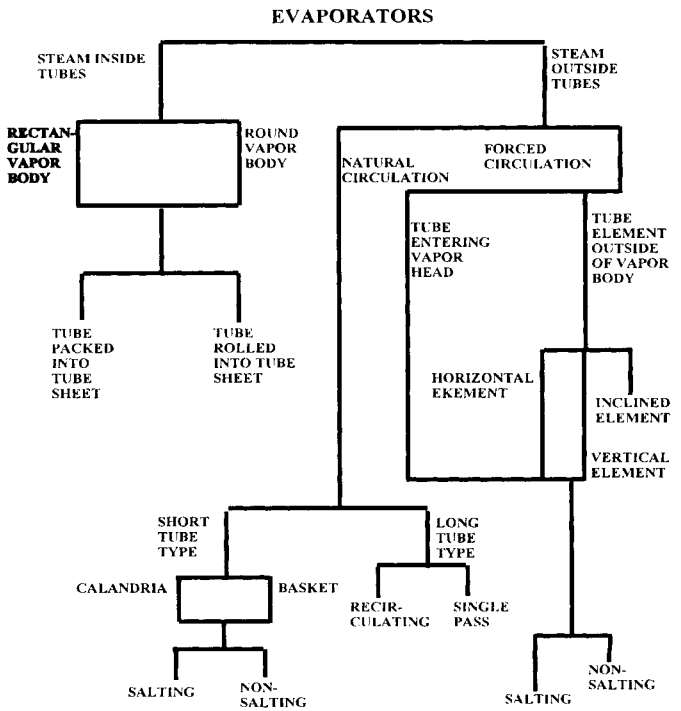

Figure 1. *Classification system for evaporating equipment.*

Major evaporator designs include forced-circulation, long-tube vertical (both rising and falling film), and calandria-type evaporators. The economics of a particular process will dictate the evaporator style and model best suited to a particular application. Forced-circulation and calandria evaporators are required for processes where crystals are formed. These evaporators are designed to keep crystals suspended in solution to prevent scaling of the equipment. Long-tube vertical evaporators are used to concentrate a liquid that does not have solids present.

Evaporators require a source of heat to operate. This heat may be supplied from a boiler, gas turbine, vapor compressor, other evaporator, or a combination of sources. Multiple effect evaporators are very popular when cheap, high pressure steam is available to heat the system. A Mechanical Vapor Recompression evaporator would use electricity or a gas turbine to drive a compressor that recycles the heat in the evaporator.

Evaporators have performed successfully in a number of industrial applications. Typical materials that are processed in evaporators include Caustic Soda, Caustic Potash, Sodium Carbonate, Sodium Dichromate, Sodium Nitrate, Ammonium Nitrate, Phosphoric Acid & Superacid, Potash, Urea, Glue, Glycerine,

Tankwater, Bright dip acid (phosphoric), Cyanide rinse bath, Pickle Liquor, Sodium Aluminate Liquor, N.S.S.C. Liquor, Kraft Liquor, Soda Liquor, Sulfite Liquor, Stillage, Corn Syrup, Gelatin, Salt, Soybean Oil, Steepwater, Sugar, Whey, Mercerizing Caustic, Nylon Salt, Rayon Spin Bath, and Sodium Sulfate.

As the name implies, the liquor in a Forced-Circulation Evaporator is pumped through the tubes to minimize tube scaling or salting when precipitates are formed during evaporation. A forced-circulation evaporator (with a submerged inlet) complete consists of a single-pass vertical heat exchanger, elutriating leg, cyclone, and top mounted barometric condenser. Slurry is pumped from the bottom cone of the vapor body through the tubes of the vertical heat exchanger, where heat is added, and back into the vapor body where evaporation occurs. Sufficient slurry height (submergence) is maintained above the tangential inlet on the vapor body and above the top tubesheet of the heat exchanger to suppress mass boiling in the inlet and prevent surface (local) boiling on the tube surface. This is necessary to preclude salt precipitation on the tangential inlet and tubes.

A high circulation rate is provided for adequate tube velocity to achieve good heat transfer. Therefore, lower slurry temperature rises are assured which minimize supersaturation of the solution. A sufficient quantity of salt crystals is suspended in the circulating system to provide seed crystals in the boiling zone for salt growth. Adherence to these basic principles of crystallization results in coarse crystals and minimal wall and tube salting, so less equipment washing is required. This conserves energy because less steam is required to boil wash water and this increases on-stream time for the evaporator. The circulating pump is usually of the axial-flow, single-elbow design, well-suited for the high flow rates and low pressure drops. These heavy duty pumps operate at low speeds, which reduce maintenance and minimize mechanical attrition of the salt crystals. Circulating piping interconnects the vapor body, the heat exchanger and the pump. Conical liquor chambers provide gradual, low pressure drop transitions from the circulating piping to the tube bundle which is particularly important for the establishment of uniform feed to the tubes. To provide for thermal expansion without expansion joints, the circulating pump base is spring-mounted. As an alternative, the entire pump may be hung from the circulating piping. The tangential inlet provides excellent mixing of slurry in the vapor body because of the circular motion it creates. Secondary vertical currents are also generated, mixing body slurry with the hotter slurry entering the vapor body to reduce the degrees of flash. This agitation minimizes salt buildup on the bottom cone of the vapor body. A swirl breaker is provided in the circulating slurry outlet. The vapor body is conservatively designed both in diameter and height. It is important to have an adequate free space above the liquor level to allow the liquor droplets entrained in the vapor leaving the boiling surface to reach equilibrium and return by gravity to the circulating slurry. The large diameters result in low vapor velocities which minimize entrainment and provide adequate retention time for salt growth. A mesh-type entrainment separator may be

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installed in the upper portion of the vapor body to reduce solids carryover to normally less than 50 parts per million parts of vapor. Other types of entrainment separators are also available.

The elutriating leg, attached to the bottom cone of the vapor body, is a convenient device for thickening the slurry it receives from the vapor body and for salt crystal washing and classification. Slurry enters the top of the leg through a unique slurry-inlet device, which improves washing efficiency by reducing agitation in the leg. Salt crystals are fluidized and washed in the leg with a portion of the feed liquor which enters the bottom cone and is distributed with a perforated, dished head. Smaller crystals are washed into the vapor body for additional growth and the larger crystals are discharged from a connection near the bottom of the leg.

A low-pressure-drop liquid cyclone is sometimes used to clarify liquor discharged from the evaporator. The driving force is the pressure drop across the circulating pump. Thickened slurry is returned through a wide-open cyclone underflow connection to the circulating piping before the pump suction.

An innovation is a direct-contact condenser mounted on the vapor body. A short piece of vertical pipe connects the vapor body with the condenser to minimize piping and pressure drop. This design also eliminates structural steel for support of a separate condenser. For cooling tower applications, the hotwell is elevated to permit gravity flow of water from the hotwell to the top of the cooling tower, thus eliminating the need for a pump.

A forced-circulation evaporator with horizontal heat exchanger and a top-mounted stripping column operates in the following manner. Reflux liquid is introduced on the top tray of the column to strip one or more compounds from the water vapor. Stripping columns are used for special applications and are provided either integral with the evaporator or as a separate column. The columns are for the recovery of valuable components from the water vapor and for the reduction of volatile pollutants. Stripping columns are usually supplied with ballast and valve trays for caprolactam and boric acid service. Spray columns have been used for the recovery of fluorine generated along with the vapor during the concentration of phosphoric acid.

Heat exchangers can be located on the discharge or suction side of the circulating pump. In some cases, both discharge and suction-side heat exchangers have been provided in the same circulating system to maximize heat-transfer surface. Multiple circulating systems are provided for the same vapor body on large evaporators. Well designed systems include special provisions to reduce tube plugging caused by salt lumps.

A variety of forced-circulation evaporator designs are available, with and without elutriating legs or cyclones. Different inlets and outlets are used on the vapor body, tailored for the particular application. For some applications, such as super-

phosphoric acid and red liquor, it is necessary to use unsubmerged-inlet evaporators to reduce the volume in the system to a minimum or to minimize foaming. Because of the higher head created by the unsubmerged inlet, it is sometimes necessary to use a mixed-flow circulating pump instead of an axial-flow pump.

Typical Configurations

A standard form of evaporator is an upright cylinder, within which a nest of upright steam tubes is placed. The horizontal steam tubes are also built into the upright cylindrical body. Instead of tubes, suitably placed coils, within which steam circulates, are installed for small-scale and batch operations. Certain evaporators have both tubes and (at a lower level) coils. The function of an evaporator is to transfer heat, mainly the latent heat of the steam from the steam side through the metal wall of the vessel to the liquor on the outer side. The resistance to the passage of the heat is not only that of the metal, which is indeed very small, which means that the heat transfer coefficient is high, but also that of the condensed steam film on the one side and a film of liquor on the other side. The condensed steam film, which is essentially hot water, has a moderate resistance to heat, partly because its thickness is low; hence, its coefficient of transfer is 2,000 or 3,000. Figures 2 through 8 provide examples of common equipment that will assist the reader in becoming familiar with the various configurations and operational modes for evaporating equipment.

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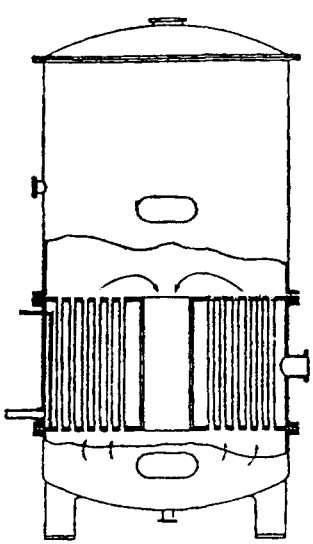

Figure 2. *Vertical evaporator.*

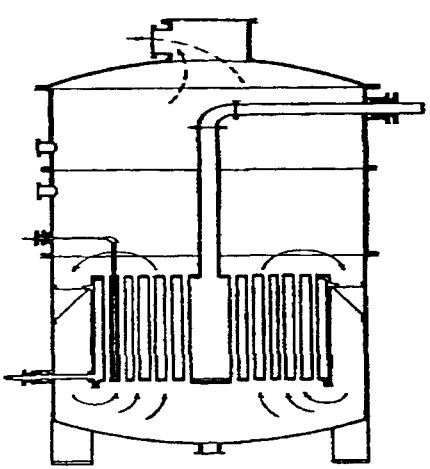

Figure 3. *Vertical long-tube evaporator.*

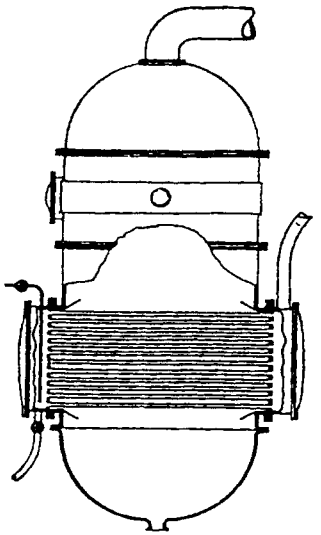

Figure 4. *Horizontal design.*

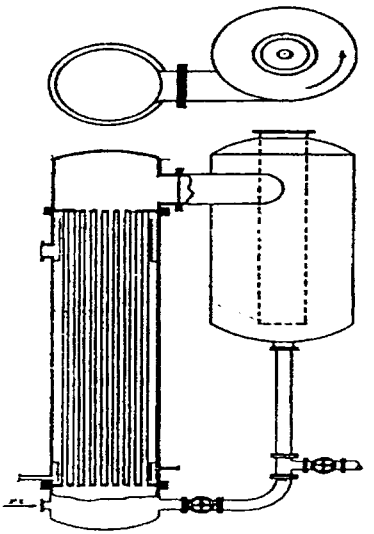

Figure 5. *Natural circulation, long tube film evaporator.*

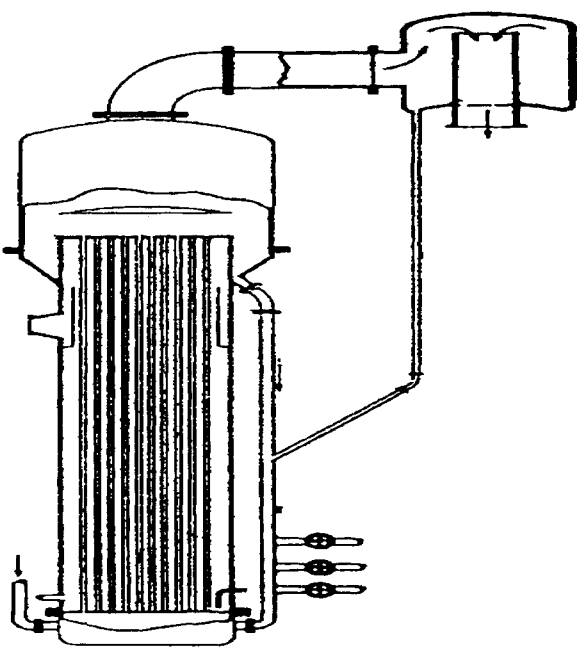

Figure 6. *Natural circulation, long tube-vertical film evaporator.*

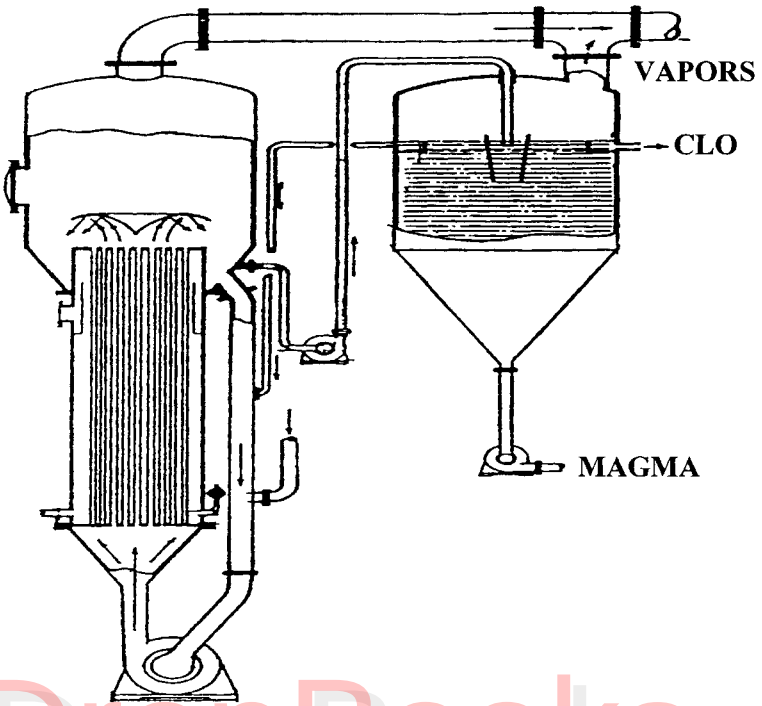

Figure 7. *Forced circulation evaporator.*

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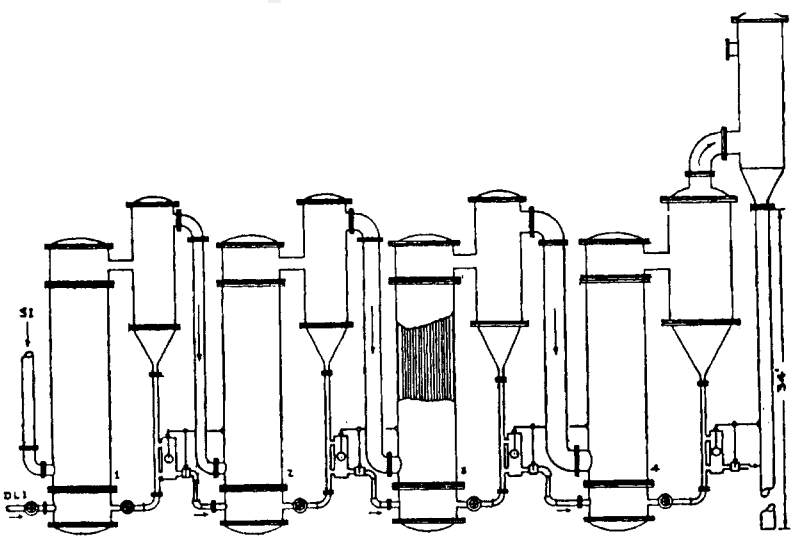

Figure 8. *Quadruple effect, natural circulation vertical tube evaporator.*

A simplified example of a Forced-Circulation Type Evaporator is illustrated in Figure 9 (in this figure the letter designations are: A. Heating element shell; B. Vapor head; C. Downtake pipe; D. Circulating pump; E. Deflector; F. Steam baffle; G. Non-condensed gas outlet; H. Condensate outlet; J. Salt settler; K. Liquor level indicator). The tubes are longer and narrower than in the standard vertical type and are contained in a heating element A between two tube sheets. This tube bundle projects part way into the vapor head B. A return pipe C leads from the vapor head to the inlet of a pump D and this pump delivers the liquid to the tubes with a positive velocity. As the liquid rises through the tubes, it becomes heated and begins to boil, so that vapor and liquid issue from the tops of the tubes at a high velocity. This material strikes against a curved deflector E that throws the liquid downward in a sheet or curtain and makes an effective separation of liquid and vapor. The steam connection is near the bottom of the tube bundle.

Just inside the shell of the tube bundle is a cylindrical baffle F that extends nearly to the top of the heating element. The steam rises between this baffle and the wall of the heating element and then flows downward around the tubes. This displaces non-condensed gases to the bottom, where they are removed at G. Condensate is removed from the bottom of the heating element at H. This evaporator is especially suited for foamy liquids, for viscous liquids, and for those liquids which tend to deposit scale or crystals on the heating surfaces. Vessel J is a salt separator.

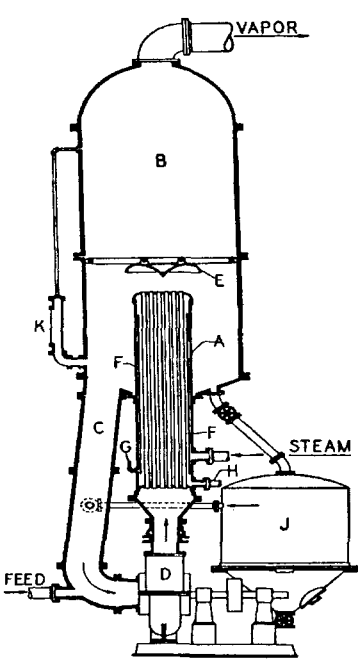

Figure 9. *Forced circulation evaporator.*

Basket Type Evaporators - the first basket - type evaporator was built in 1877 and an example of this type is shown in Figure 10. In this evaporator the liquid is inside the tubes, the steam is outside the tubes, but the downstage instead of being central is annular. In this construction the entire heating element is a single unit which may be removed bodily for repairs. The evaporator has a conical bottom, but flat bottoms are also common in this type.

The tube proportions in the basket evaporator are about the same as in the standard vertical. One important feature of the basket-type evaporator is the ease with which a deflector may be added in order to reduce entrainment from spouting. The boiling in the vertical-tube evaporator is quite violent, and this tends to cause entrainment losses. This condition is accentuated if the liquor level in the evaporator is low. A baffle such as shown in Figure 9 largely prevents these losses and is much more easily added to the basket type than to the standard type. Other differences between the standard and the basket type are largely details of construction.

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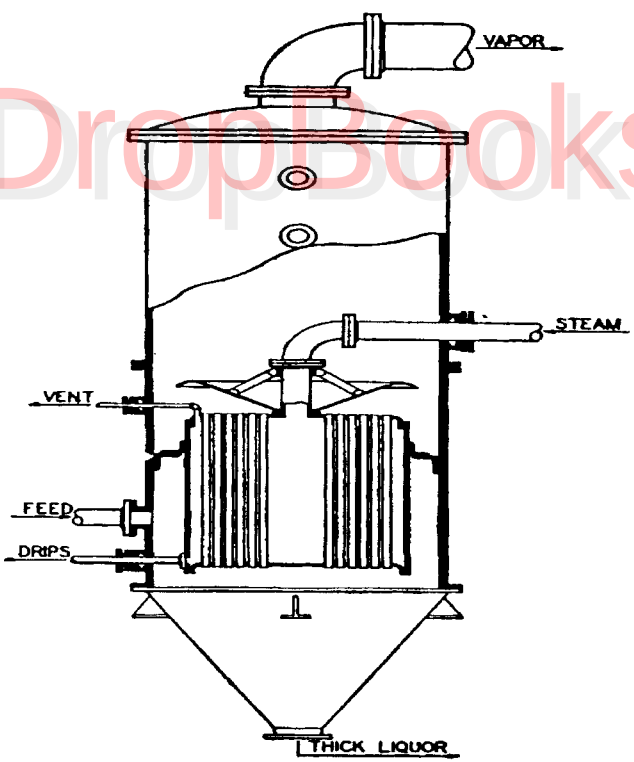

Figure 10. *Basket type vertical tube evaporator.*

Evaporators, Vertical - Tube Type - An example is shown in Figure 11. This evaporator is characterized by tube sheets A extending across the body, and a central downstage B. Tubes are rolled between these two tube sheets, and steam is introduced as shown, so that the liquor is inside the tubes and the steam is outside them. As the liquor boils, it spouts up through the tubes and returns through the central downstage. Condensate is removed from any convenient place on the bottom tube sheet such as C, and non-condensed gases are usually vented from somewhere near the upper tube sheet as at D. The exact position of feed E and discharge F is variable. There are many variations of this design; a conical bottom may be used instead of a flat bottom; and the proportions of tube length to tube diameter, ratios of body height to tube length, and other details are often modified. The first vertical - tube evaporators were built without a downtake. These were never satisfactory and the central downstage appeared very early. There have been many changes over the years, such as downtakes of different cross-section, downtakes located eccentrically, a number of scattered downtakes instead of one central one, downstage pipes entirely external to the body of the evaporator, and others.

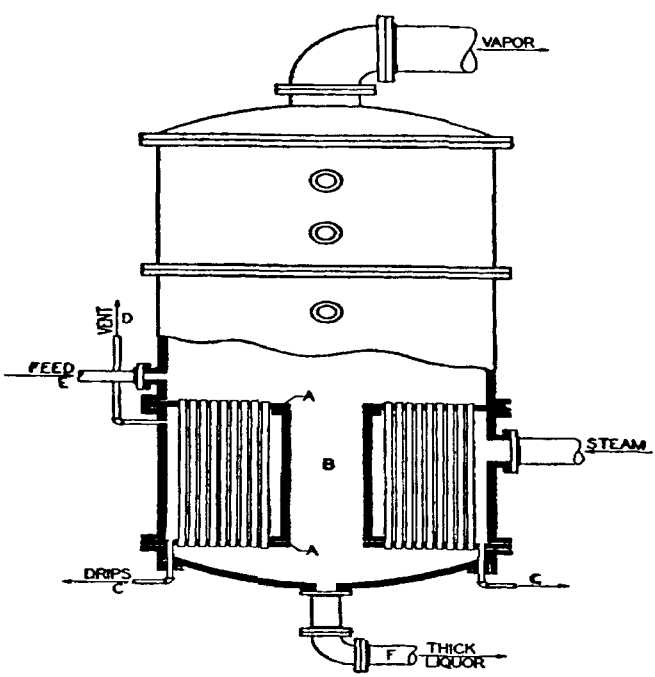

Figure 11. *Vertical tube evaporator: (A) Tube sheets; (B) Downtake; (C) Condensate outlet; (D) Non-condensed gas outlet; (E) Liquor inlet; (F) Thick liquor outlet.*

Evaporators, Horizontal-Tube Type - The basic horizontal-tube evaporator is illustrated in Figure 12. The body of this evaporator is the liquor compartment and is in the form of a vertical cylinder. It is closed, top and bottom, with dished heads, although the bottom may be conical. The lower body ring is provided on opposite sides with steam compartments, closed on the outside by cover plates and on the inside by tube sheets. Between these tube sheets are fastened a number of horizontal tubes. The two steam chests with their connecting tubes form the steam compartment, and the tube wall heating surface. Steam is introduced into one steam chest and as it flows through the tubes it washes non-condensed gases and condensate ahead of it, so that these are withdrawn from the opposite steam chest.

In ordinary operation only condensate and non-condensed gases are removed from the exit steam chest. The connection for feeding the liquid to be evaporated may be attached to the body at any convenient point (D), but the discharge for thick liquor is usually in the center of the bottom (E). Suitable brackets are cast on the bottom to rest on the supporting steel. Most evaporators are provided with sight glasses.

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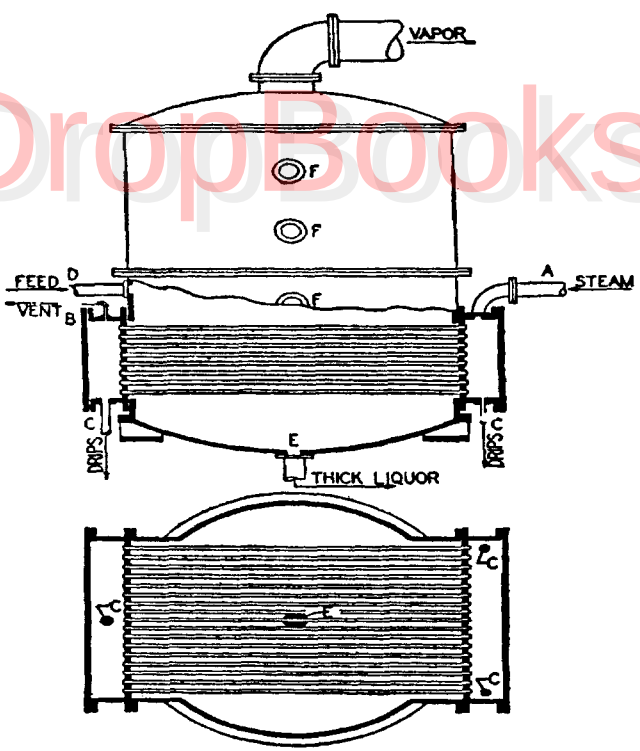

Figure 12. *Horizontal tube evaporator: (A) Steam inlet; (B) Vent for non-condensed gas; (C) Condensate outlet; (D) Liquor inlet; (E) Liquor outlet; (F) Sight glass; (G) Vapor outlet.*

There are many modifications of the horizontal-tube evaporator, but these consist largely of changes in the shape of the body castings and not at all in the general arrangement or interrelationship of the parts. The horizontal-tube evaporator is best suited for non-viscous solutions that do not deposit scale or crystals on evaporation. Its first cost per square foot of heating surface is usually less than that of the other types of evaporators.

A common design involves steam outside the tube which is shown in Figure 13. This evaporator consists of a cylindrical shell A closed by tube sheets B at either end. The shell may be from 8 to 20 ft. long. A number of tubes are fastened in these tube sheets by rolling. This term refers to a method of fastening a tube in a tube sheet by means of a tool containing rollers and inserted into the end of the tube. The rollers are so mounted that as the tool is rotated the tube is gradually expanded until it makes a tight fit with the tube sheet. Over these tube sheets are fitted distributing plates which have return bends cast in them in such a way that the tubes are connected in series in groups of three or five. One of these distribution plates carries feed connections that register with the end of the first tube of each group. The other distribution plate has openings opposite the other end of each group. Around this end is built a large separating chamber D provided with baffles E. The shell is provided with an inlet connection for steam F and outlets for noncondensed gases at H and condensate at G.

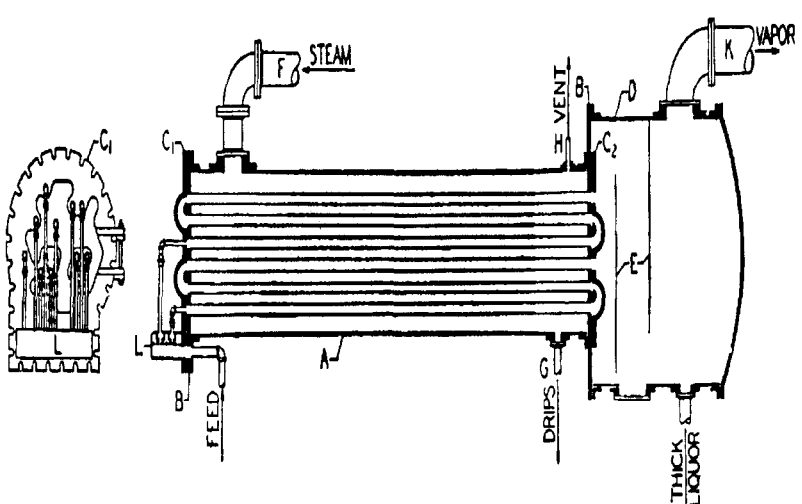

Figure 13. *Steam outside tube evaporator: (A) Shell; (B) Tube sheets: (C_1, C_2) Distributing plates; (D) Vapor head; (E) Baffles; (F) Steam inlet; (G) Condensate outlet; (H) Non-condensed gas vent; (J) Thick liquor outlet; (K) Vapor outlet; (L) Liquor feed box.*

As the liquid is introduced into the feed end of each group of tubes, it begins to boil, and the volume of vapor so formed is extremely large as compared to the volume of liquid from which it is formed. As the liquid passes down the tube, therefore, the combined volume of steam and liquid becomes larger, the velocity is correspondingly greater, and a mixture of steam and spray issues from the end of each coil and strikes against the baffles that serve to separate vapor from liquid. The concentrated liquid is drawn off at the bottom of the separating head at J, and the vapor is removed from the top at K. The liquor is not returned from J to the feed end but passes through the coils only once. All of the feed pipes are usually connected to a common feed box L, which is cast as a part of the lower edge of the feed-distributing plate. The individual feed lines extending from this box to the various coils contain throttle valves or orifices to distribute the feed evenly over the various coils.

Solvent evaporators are a workhorse in the CPI. In a single-effect, LTV rising-film evaporator evaporation occurs inside the evaporator tubes, so it is used primarily to concentrate non-salting liquors. To provide for good heat-transfer rates, the Delta T (i.e., temperature difference) between the heat-transfer medium and the liquor should be greater than 15 °F, and preferably greater than 20 °F. Tubes are normally 3/4" to 2" in diameter and from 10 to 30 feet long. Operation of the rising-film evaporator is straightforward. Liquor is fed into the bottom liquor chamber and then into the tubes. There it is heated with condensing steam or any other suitable heat-transfer medium, such as Dowtherm or hot liquor. If the vapor pressure of the feed equals or exceeds the system pressure at the bottom tubesheet, vaporization will occur immediately. For colder feed, the lower portion of the tubes is used to preheat the liquor to its boiling point. Vaporization then begins at that height within the tubes where the vapor pressure of the feed liquor equals the system pressure. As the liquor climbs up the inside of the tubes, additional vapor is generated and the velocity of the liquid-vapor mixture increases to a maximum at the tube exit. The outlet mixture impinges upon a deflector, mounted above the top tubesheet of the heat exchanger, where gross, initial separation of the liquid from the vapor occurs. Additional liquor is separated from the vapor by gravity as the vapor rises in the vapor body. A mesh-type or centrifugal entrainment separator can be installed near the top of the vapor body to remove most of the remaining traces of liquid from the vapor. The exit vapor is conducted to the next effect of a multiple-effect evaporator, to a compressor or to a condenser.

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Heat-transfer rates are enhanced in the non-boiling section by surface or local boiling and in the boiling section by nucleate boiling. As expected, the heat-transfer rates in the boiling zone are several times greater than those in the non-boiling zone, so it is important to reduce the non-boiling zone to a minimum.
Different two-phase flow schemes are created in the boiling zone, including *slug flow*, where a slug of liquor is followed by a slug of vapor, similar to the perking in a coffee percolator; *annular flow*, where a ring of liquor encases a center core

of vapor and liquid mist; and *mist flow*, where vapor blankets the tube surface. Mist flow should be avoided because poor heat transfer results when there is not enough liquid present to wet the tube walls. To avoid mist flow, it is sometimes necessary to recycle concentrated product from the vapor body to the bottom liquor chamber so as to supplement the feed liquor.The LTV rising-film evaporator can also be adapted to concentrate moderate scaling liquors when a pump is used to recycle concentrated product from the vapor body to the bottom liquor chamber. This is known as forced recirculation. The vapor body is integral with the heat exchanger. When the heat exchanger is too large to use the integral configuration, when quicker access to the tubes is desired for maintenance, or when surge volume in the vapor body is required for level control, the vapor body is separated from the heat exchanger. A skirt-type baffle replaces the deflector as the initial separator.

In a falling-film evaporator with separate vapor body and heat exchanger, liquor is fed into the top liquor chamber of the heat exchanger where it is distributed to each tube. The liquor accelerates in velocity as it descends inside the tubes because of the gravity and drag of the vapor generated by boiling. Liquid is separated from the vapor in the bottom liquor chamber of the heat exchanger and with a skirt-type baffle in the vapor body. A supplemental entrainment separator can be installed in the upper portion of the vapor body to reduce liquid entrained with the vapor to a minimum. A direct-contact condenser is used to condense the vapor with water. Concentrated liquor is discharged from the bottom liquor chamber and cone bottom of the vapor body. The vapor body can also be provided as an integral component of the heat exchanger, except the heat exchanger would be located above the vapor body in the falling-film configuration. Evaporation also occurs inside the tubes of the falling-film evaporator. The unit can be used to concentrate the same non-salting liquids concentrated in rising-film evaporators, and it is suitable for concentrating more viscous liquors. Tube sizes and lengths are comparable to those for rising-film evaporators. The falling-film evaporator is particularly useful in applications where the driving force in temperature difference between the heat-transfer medium and the liquid is small (Delta-T's of less than 15 ^{0}F). The retention time for liquor in this evaporator is less than that for a rising-film evaporator. The combination of short liquid retention time and the ability to operate at a low Delta-T makes the falling-film evaporator ideal for concentrating the most heat-sensitive materials.
High heat-transfer coefficients are attained in falling-film evaporators when a continuous film of liquid, preferably at its boiling point, flows down the inside tube wall with a vapor core in the tube center. For some applications, however, it is necessary to supplement an insufficient quantity of feed liquor with product liquor pumped to the top liquor chamber to avoid vapor blanketing of the inside tube surface.
The *Calandria Evaporator* is applied less often today than it was years ago. For most applications, however, the lower equipment costs for other designs has prompted the replacement of calandria evaporators with rising-film, LTV falling-

film and forced-circulation evaporators. The *Calandria Evaporator* has a heat exchanger (with tubes usually less than six feet long) integral with the vapor body. The level is maintained in the upper portion of the tubes or above the top tubesheet and the circulation pattern is up through the tubes and down through a central pipe called a "downcomer". Circulation is created by the difference in specific gravity between the body liquor and the heated liquor and vapor generated inside the tubes, plus a vapor lift effect. The *Calandria Evaporator* can be used for salting-type applications; however, an agitator located inside or beneath the downcomer is recommended to suspend the salt crystals in the lower portion of the body. Although the agitator creates some flow through the tubes, most of the flow is still created by "thermo-syphon". The calandria is also used for batch-type concentration of liquors.

Applications and Process Schemes

There are numerous applications in solvent recovery processes where evaporation equipment are employed. Figure 14 provides an example of a process scheme for toluene-di-isocyanate recovery. This is an example of continuous vacuum evaporation of distillation residues.

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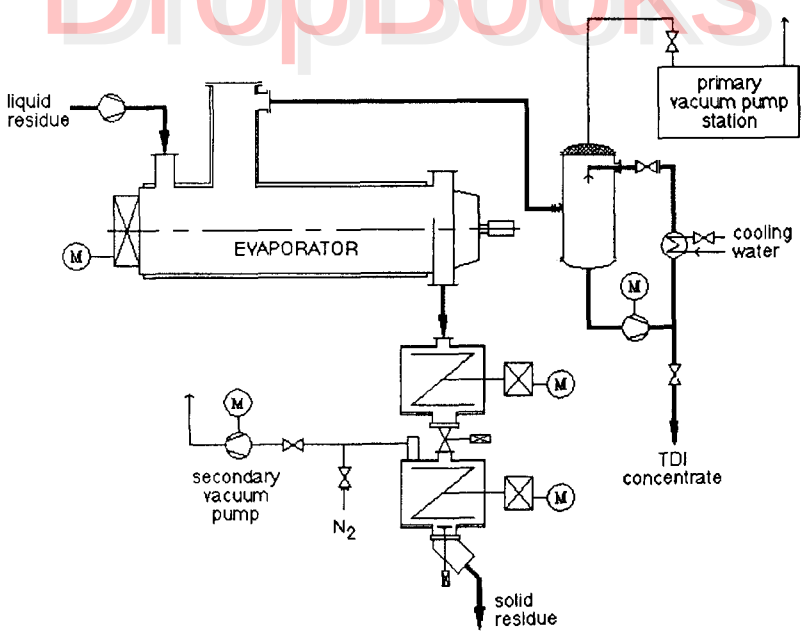

Figure 14. *Process flow scheme for TDI recovery by evaporation.*

The vaporized solvent is recovered by condensation. The remaining non-toxic waste typically is free-flowing powder with residual solvent contents that are below 0.5 %. This is an economical process scheme involving only a single-step process for the complete recovery of the solvent. The design is a closed system, which makes it environmentally safe, and yields a non-toxic solid residue for disposal. Other applications for the process configuration illustrated in Figure 14 include handling pasty/sticky residue with toxic-high boiling solvents from chemical or pharmaceutical productions, and the preparation of washing solutions.

Another example of a solvent recovery process is that of DMAC (Dimethyl-acetamide) recovery, illustrated in Figure 15. In this example, the solvent recovery involves continuous evaporation under vacuum. The vaporized solvent exits through a dust filter and is recovered for reuse by condensation. The dried residue can be either a free flowing solid or in the form of a viscous product. The residual solvent content is typically reduced to a level of about 0.1 % in this process. The process flow scheme illustrated in Figure 15 provides for almost complete recovery of a solvent in on step, and thereby minimizes both disposal costs and nw solvent purchase costs. Systems are capable of handling very difficult materials ranging from highly viscous ands sticky products to crust-forming solid-like materials. Other typical examples are in the reprocessing of off-spec polymer and in processing paint sludge

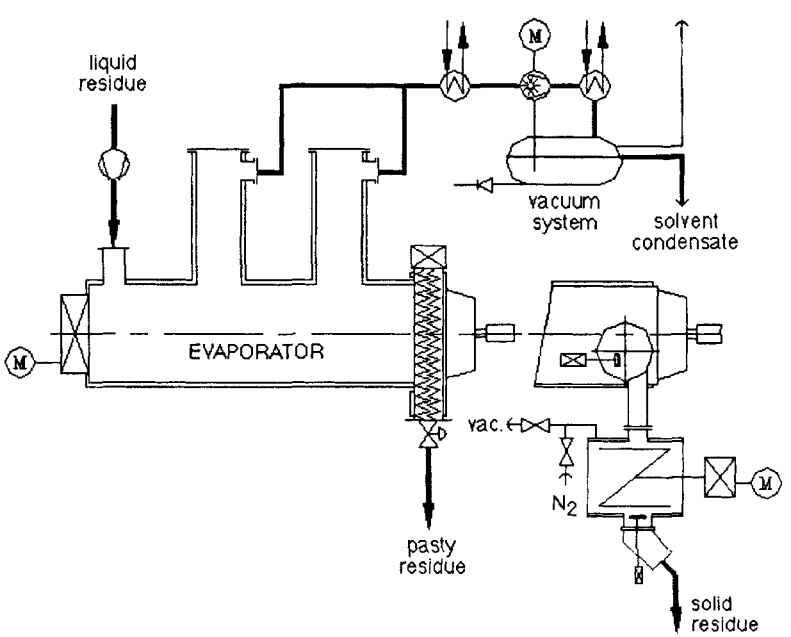

Figure 15. *Process flow scheme for DMAC recovery by evaporation.*

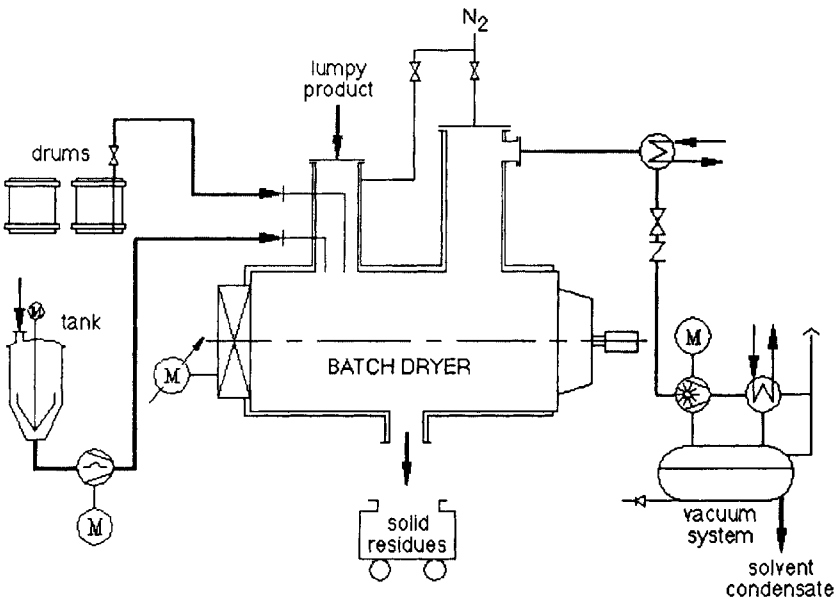

Figure 16. *Process flow scheme for a batch vacuum drying operation in the processing of rubber wastes.*

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An example of continuous evaporation is in the processing of synthetic rubber solutions containing EPDM, SBr, Br, and the like. In the continuous evaporation of synthetic rubbers, a metered solution is passed through a preheater. The heated solution is then flashed into an evaporator where the remaining solvent is evaporated upon contact by heat. The viscous rubber is then discharged through a twin screw extruder, which serves for final stage drying of the rubber product. The example, shown in Figure 16, can be describes as direct, single-stage devolatization. This process has an advantage over more conventional rubber finishing (i.e., drying and solvent recovery) operations. More conventional systems use a combination of steam stripping, expeller/expanding/drying equipment and steps, as opposed to the single-step desolventizing process illustrated in Figure 16. As such, there are both energy savings, as well as environmental credits with a more compact, closed system. Typical applications include the processing of atactic polypropylene, elastomers, adhesives, silicon mass, and ABS/ATS polymers.

Another example in the polymers industry is illustrated in Figure 17, which is a process aimed at the batch drying of waste residue with solvent recovery. In this application liquid or viscous waste solutions are pumped into a batch dryer where they are dried under vacuum to a solid granular residue. Vaporized water and solvent are recovered by condensation and then separated by gravity. The process scheme is flexible, offering a range of temperatures and vacuum levels for treating

very sticky and crusty residues. It provides nearly complete solvent recovery and yields inert solid residues, thereby minimizing air pollution emissions. Chemical conditioning of the residue also occurs in the process. Typical applications can include the preparation of washing solutions, processing of paint residues, process residues, oil sludges, and contaminated oil sludges.

An example of the use of evaporating equipment in the food processing industry is illustrated in Figure18. This is an example of continuous vacuum drying to produce milk crumb. Preconcentrated liquid material is continuously crystallized and dried through a viscous, pasty intermediate phase until a crumbly product is discharged via a piston lock system. Maillard-type reactions impart special flavoring to the final product. Exact temperature control produces a high-quality, good tasting product. The compact design for this system features a clean-in-place system that requires little floor space. The process provides safe processing during sticky, crust-forming, intermediate stages of product processing.

The system shown is heavily used in food processing applications such as milk or chocolate crumb production, sugar substitutes, modified starch, and alginates. In addition to food processing applications, such a system is used in the processing of heat-sensitive pharmaceuticals, polymer suspensions like latex, in processing pigments and dyestuffs, and pesticides.

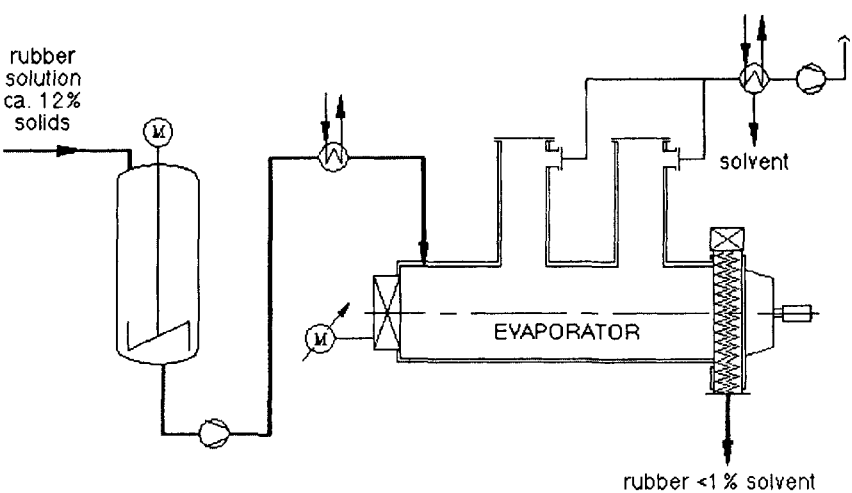

Figure 17. *Process flow scheme for continuous evaporation in a rubber desolventizing process.*

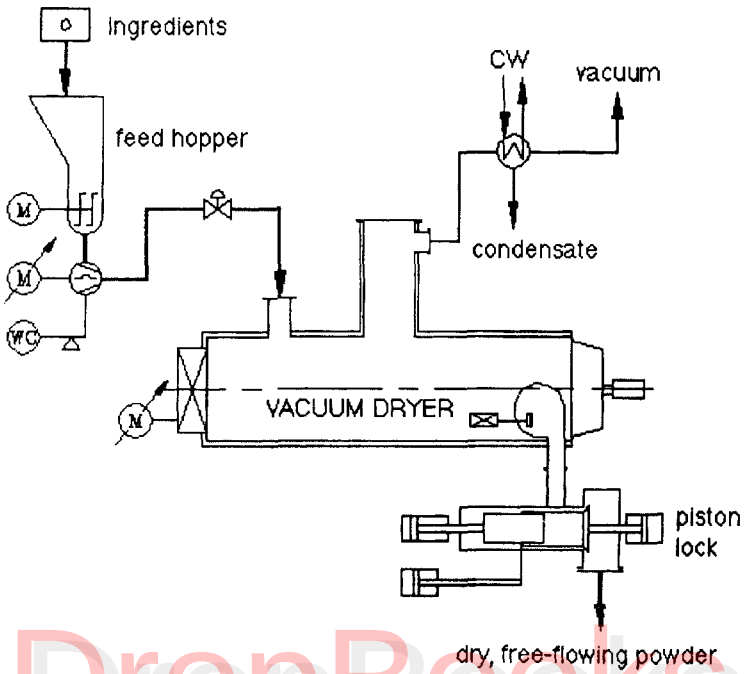

Figure 18. *Process flow scheme for continuous vacuum drying in the food processing industry.*

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An example of continuous contact drying of residues and sludge is shown in Figure 19. This process is designed for handling paint sludge. The sludge is homogenized and metered to the continuous dryer by a paste hopper-feed screw system, or by special piston pumps. The residue is continuously dried through a viscous paste phase in a single step without recycling.

The resulting dried residue is continuously discharged through a twin screw or rotary valve. Solvent and vapors are condensed and recovered. The system shown is versatile and can process washing and cleaning solutions, sticky residues from chemical, pharmaceutical and food processing applications, contaminated soils, and oil sludge, in addition to paint sludge.

An example in the pharmaceutical industry is shown in Figure 20, which is a batch contact drying application. In drying pharmaceuticals. The filter cake is batch dried at low temperatures under vacuum to a free flowing product. Solvent vapors leave through a dust filter. The solvent condensate passes through an indirectly-cooled, solvent ring-pump, and is recovered. Additional applications include the processing of pigments and dyestuffs, temperature sensitive pharmaceuticals and foods, polymers and additives, various residues, sludge and salts.

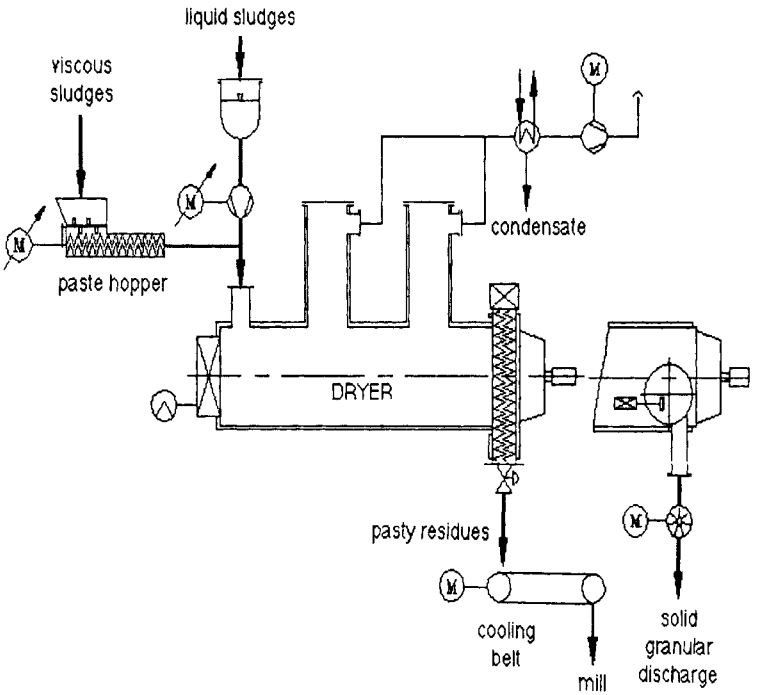

Figure 19. *Process flow scheme for continuous contact drying of residues and sludges.*

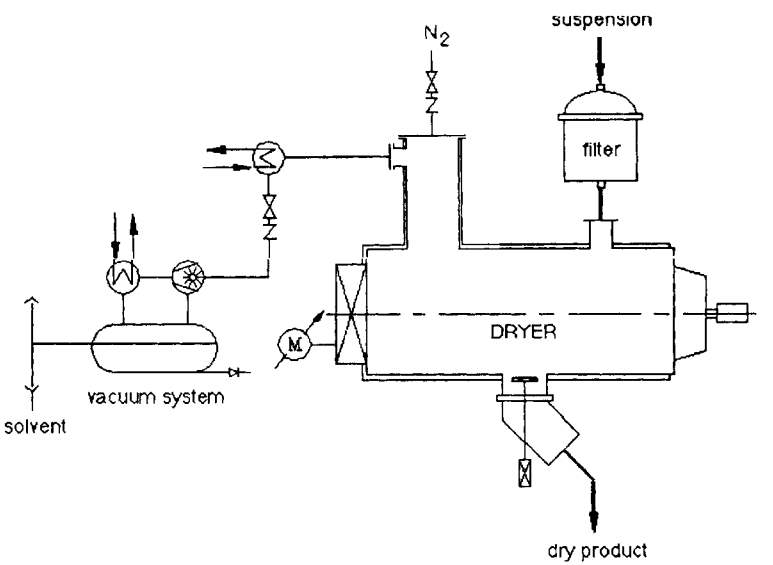

Figure 20. *Process flow scheme for batch contact drying application.*

Energy Requirements and Multiple Effects

The change in free energy that is required in evaporation is a measure of the theoretical difficulty of carrying out any reaction or separation, which is the minimum amount of work necessary to effect the change involved, if performed under theoretically perfect and reversible conditions. Any process should be able to operate with a work consumption that does not greatly exceed this value. It is important to recognize that this energy must be supplied as work. In other words, heat in itself is not the equivalent of energy, although for convenience energy is often expressed in heat units. From thermodynamic considerations it can be shown that the minimum amount of work required is very small, amounting, in the case of a 10 per cent water solution of glycerin, to a mere 6.4 Btu per lb of water evaporated. Obviously the more concentrated the solution, the greater is the theoretical energy required, but even for concentrated solutions this value remains relatively small. For colloidal solutions, i.e., those of high molecular weight, the reversible work is even less. Theoretically, therefore, the greater part of the heat supplied in evaporation should be recoverable, and such is the case in real life.

Since the free energy necessary is work and not a mere heat requirement, available heat engines actually convert 25 per cent or more of the heat supplied into work. Hence, the theoretical heat requirement for a 10 per cent glycerin solution is only 32 Btu per lb of water evaporated. From this point of view, a simple evaporator is a very inefficient system. However, the heat required for evaporation is almost entirely converted into latent heat of vaporization, and is, in this form, carried out of the evaporator by the vapor and can be again utilized by condensing the vapor. A simple example of this is a plant operation where a certain amount of water is being evaporated at atmospheric pressure in the process of concentration of an aqueous solution. Most of the heat consumed appears as latent heat in the water vapor. It could not, however, readily and directly be employed to evaporate more water under these conditions, since on condensing it this heat is evolved at the boiling point (100°C.). As such, no temperature difference exists to force the heat through the necessary heating surface into the fresh solution to be evaporated. If, however, the same plant operation had a corresponding supply of alcohol to be distilled, the water vapor from the evaporator could readily be used as a heat supply therefore, by condensation in the heating coils of the alcohol still, rather than by using boiler steam for the still.

If, in turn, this same plant had ether to be distilled, the heating coils of the ether still could be used as a condenser for the alcohol vapor, thus avoiding the use of boiler steam in the ether still. In this way a succession of liquids of progressively lower boiling points can be evaporated with a single supply of heat. In other words, in the process of evaporation, the heat used is merely degraded in temperature and this degradation is for all practical purposes slight.

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This leads us to a discussion on multiple effects. The above conditions enable a given quantity of heat to be used for evaporation a multiple of times by boiling a series of liquids of successively lower boiling points. This is a very practical application throughout the CPI. It is important to note that the boiling point of a water solution depends upon the pressure above it, and by a careful selection of pressures a solution can be made to boil at the temperatures given for alcohol and ether or any more suitable temperatures, and a similar economy of heat can be effected. Each evaporator is referred to as an effect, and the overall process is called multiple-effect evaporation. We consider the classic example given by Walker, Lewis, McAdams and Gilliland (1937), shown in Figure 21, whereby the steam entering the coils of effect A is saturated exhaust steam at 2.5 lb. gauge (17.2 lb. per sq. in. abs. pressure) and at 220 °F, and the pressure in the vapor space of A corresponds to a vacuum of 8.75 in. of mercury, or with normal barometer 10.4 lb per sq.in.(abs) pressure. The water in A will boil at 195 °F and furnish steam to the heating coils of B at this temperature. The vacuum in the vapor space of B is 19.1 in. and hence water will boil at 165 °F. The vapor coming from B and entering the heating coils of C at 165 °F will cause water to boil in C at 125 °F where the vacuum is 26 in. Vapor from C at 125 °F will then pass to the condenser. Since the heating element of A is above atmospheric pressure, the condensed water in it can flow from it by gravity at a little below 220 °F. But the condensed water in the heating space of B is under vacuum; hence the pump P' must be used to exhaust the heating space in B and deliver the water at atmospheric pressure. Similarly, the condensate in the heating space of C must be pumped out. The condensed water from the heater in B at about 195 °F is in position to give up part of its sensible heat to the liquid boiling in C at 125 °F, and by leading it into the heating element of C part of it flashes into steam due to the lower pressure, thereby reducing its temperature to 165 °F (the temperature of the steam in the coils of C).

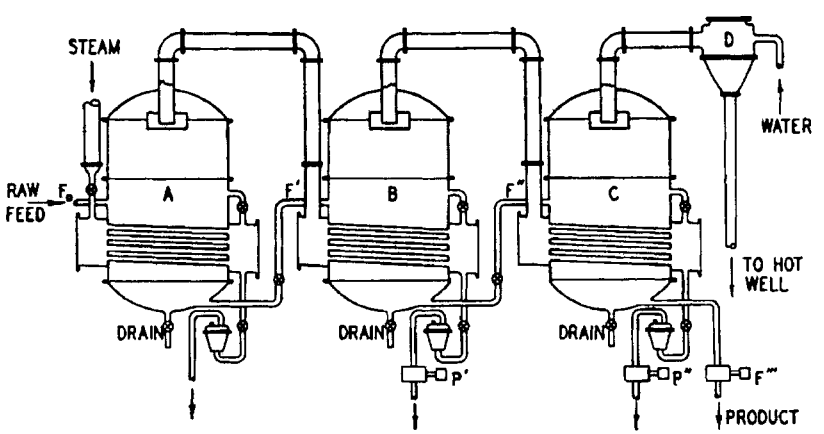

Figure 21. *Multiple effect evaporator.*

The hot condensate from the heater in A can be returned to the boiler. The solution to be evaporated is fed at 195 °F into the liquor spaces of effect A at F_o by means of a pump or by gravity. Since the pressures in B and C are progressively lower, the solution can be drawn in sequence through the system by the pipes F' and F" and finally lifted to atmospheric pressure by the pump F''. This arrangement is called parallel flow, or forward flow. In order to permit the removal of fixed gases from the heating space of all the effects, each is connected by a small pipe to the vacuum condenser or to the body of the vapor space of the same effect which is obviously at lower pressure. Where the noncondensable gas is small, the latter is better practice, as this, method acts as a safeguard against the loss of steam due to its discharge with fixed gases.

In the above example, 1 lb of initial steam should evaporate approximately 1 lb of water in each of the effects A, B and C. In practice however, the evaporation per pound of initial steam, even for a fixed number of effects operated in series, varies widely with conditions, and is best predicted by means of a heat balance.This brings us to the term *heat economy*. The heat economy of such a system must not be confused with the evaporative capacity of one of the effects. If operated with steam at 220 °F in the heating space and 26 in. vacuum in its vapor space, effect A will evaporate as much water (nearly) as all three effects costing nearly three times its much; but it will require approximately three times as much steam and cooling water. The capacity of one or more effects in series is directly proportional to the difference between the condensing temperature of the steam supplied, and the temperature of the boiling solution in the last effect, but also to the overall coefficient of heat transfer from steam to solution. If these factors remain constant, the capacity of one effect is the same as a combination of three effects.

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It is important to note that the highest temperature employed in an effect is limited by the corresponding pressure, which must not exceed the working strength of the apparatus. At times it is, of course, limited by the pressure of the available steam supply, and also by the temperature to which the solution may be safely heated (i.e., without destroying its properties). The lowest pressure that can be realized depends upon the temperature of the cooling water. These limiting temperatures, together with the nature of the solution to be evaporated and the product required, are factors not usually within the control of the designer of evaporative equipment, but are already determined. Collectively, they are designated the "terminal conditions" of an evaporative system, and represent the fixed and accepted limitations under which the equipment must be designed to operate.

Finally, we may note that every solution exerts a vapor pressure less than that of the pure solvent at the same temperature. Corresponding to this vapor-pressure lowering is an equivalent boiling-point raising. Dissolved substances lower the vapor pressure of the solvent. Such reduction increases with the concentration of solute. Since a solution boils when its vapor pressure reaches that of its surroundings, it must be heated to a temperature above the boiling point of the pure

solvent before ebullition can take place. The vapor rising from a solution is therefore superheated by an amount equal to this boiling-point raising (b.p.r.) and it must be cooled by this amount before it will condense and give up its heat of vaporization. While the amount of superheat is small when compared to the latent heat of the vapor, it should be noted that this latent heat of vaporization becomes available for use only when the vapor is cooled to a temperature lower than that of the solution from which it came by the amount of this b.p.r. This loss of temperature difference from condensing steam to boiling liquid in the next effect increases the heating surface required for the transfer of the heat.

Flow Regimes

In gas-liquid flow the two phases can be distributed in the equipment in many configurations called flow regimes or flow patterns, differing from each other in the spatial distribution of the interface. The flow pattern depends on the operational variables, physical properties of the fluids, and geometrical variables of the system. Hydrodynamics of the flow, as well as the mechanisms of momentum, heat and mass transfer, change significantly from one flow regime to another. As a result, accurate understanding of any process in two-phase flow depends on the knowledge of the existing flow patterns, and determining the flow pattern is the first step for developing techniques to accurately predict pressure drop, heat and mass transfer, gas and liquid holdup, etc. - the basic parameters needed in sizing evaporators. The flows observed may take many different configurations with respect to the distribution of the gas-liquid interface. Transitions between the various configurations are usually gradual with respect to the flow rate changes or the pipe inclination. The main task in classifying the flow patterns is to group together flow configurations that have common character according to some defined definitions. The designation of the flow pattern should be based on the flow configuration that has basically the same character, pertaining to the distribution of the interfaces and the mechanisms dominating pressure drop and heat and mass transfer. We will consider only the most standard flow regimes encountered in evaporative systems, and examine them only from the standpoint of simply geometry flows, such as in pipes. Typical flow regimes encountered in horizontal pipe flow configurations are summarized in Figure 22. A brief description of these follows:

Stratified (S) - Liquid flows at the bottom of the pipe with gas at the top. The stratified pattern is subdivided into stratified smooth (SS) where the liquid surface is smooth, and stratified wavy (SW) where the interface is wavy.

Intermittent (I) - In this flow pattern the liquid inventory in the pipe is non-uniformly distributed axially. Plugs or slugs of liquid that fill the pipe are separated by gas zones that contain a stratified liquid layer flowing along the bottom of the

pipe. The liquid may be aerated by small bubbles, which are concentrated at the front of the liquid slug and the top of the pipe. The intermittent pattern is usually subdivided into slug and elongated bubble flow patterns. The elongated bubble pattern is considered the limiting case of slug flow when the liquid slug is free of entrained gas bubbles. With this criterion, the elongated bubble-slug transition is quite sharp and easy to identify.

Annular (A) - The liquid flows as a film around the pipe wall. A liquid film surrounds a core of high velocity gas that may contain entrained liquid droplets. The film at the bottom is normally thicker than at the top depending upon the flow rates of the liquid and gas phases. At low gas rates at which a transition to annular flow from slug flow is generally observed, most of the liquid flows at the bottom of the pipe. The upper walls are intermittently wetted by large aerated, unstable waves that sweep through the pipe. This is not slug flow, which requires a competent bridge of liquid, nor is it considered to be fully developed annular flow, which requires a stable film over the entire pipe perimeter. It is therefore designated as the wavy annular pattern, or sometimes called "proto-slug" flow.

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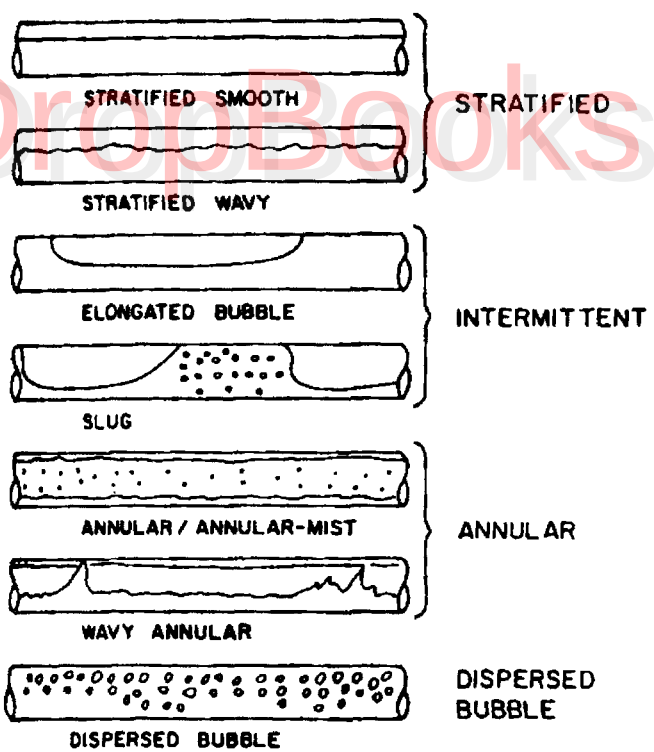

Figure 22. *Flow patterns in horizontal flow.*

Dispersed bubble (DB) - The gas phase is distributed as discrete bubbles within a continuous liquid phase. At the transition boundary most bubbles are located near the top. As the liquid rate increases, the bubbles are more uniformly dispersed.

The flow patterns typically encountered in vertical pipe flow are illustrated in Figure 23. The types of flow patterns encountered are as follows:

Bubble flow - The gas is roughly uniformly distributed in the form of small discrete bubbles in a continuous liquid phase. The flow pattern is designated as bubble flow (B) at low liquid flowrates, and as dispersed bubble (DB) at high liquid flow rates in which case the bubbles are finely dispersed in the liquid.

Slug flow (SL) - Most of the gas is located in large bullet-shaped bubbles that have a diameter close to that of the pipe diameter, and move upward. The bubbles are separated by slugs of continuous liquid that bridges the pipe and usually contains small gas bubbles. Between the bubbles and the pipe wall, liquid falls downwards in the form of a thin falling film. When the flow is low there appears well-defined gas-liquid boundaries, and the liquid slug is free of bubbles. This case is often referred to as plug flow. When the flow is faster, boundaries are not very clearly discerned and froth is generated, and the term slug flow is used.

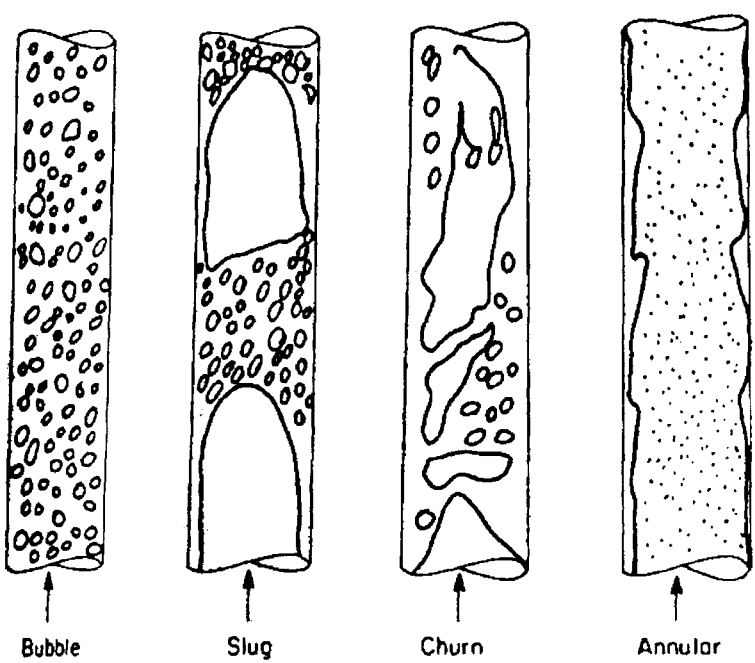

Figure 23. *Flow patterns in vertical pipe flow.*

Churn flow (CH) - This is similar to slug flow, but is much more chaotic, frothy, and disordered. The bullet-shaped bubble becomes narrow and its shape is distorted. The continuity of the liquid in the slug between successive bubbles is destroyed by a high gas concentration in the slug. As this happens, the liquid the liquid slug falls. This liquid accumulates, forming a bridge, and is again lifted by the gas. Characteristic of churn flow is an oscillatory motion of the liquid slug. For higher rates the flow is even more disordered and is sometimes called froth flow. Annular flow (A) - The flow is characterized by the continuity of the gas phase along the pipe in the core. Liquid phase moves upwards partially in the form of drops entrained in the gas core. The annular flow is termed wispy annular when the entrained phase is in the form of large lumps or "wisps". The terms froth, mist, or semi-annular describe conditions between churn and annular.

In vertical downward flow as well as in upward and downward inclined flows, the flow patterns that can be observed are essentially similar to those described above, and the definitions used can be applied. Experimental data on flow patterns and the transition boundaries are usually mapped on a two dimensional plot. Two basic types of coordinates are generally used for this mapping - one that uses dimensional coordinates such as superficial velocities, mass superficial velocities, or momentum flux; and another that uses dimensionless coordinates in which some kind of dimensionless groups are used as coordinates. The dimensional coordinates maps are inherently limited to the range of data and flow conditions under which the experiments were conducted. In spite of this limitation, it is widely used because of its simplicity and ease of use. Figure 24 provides an example of such a map.

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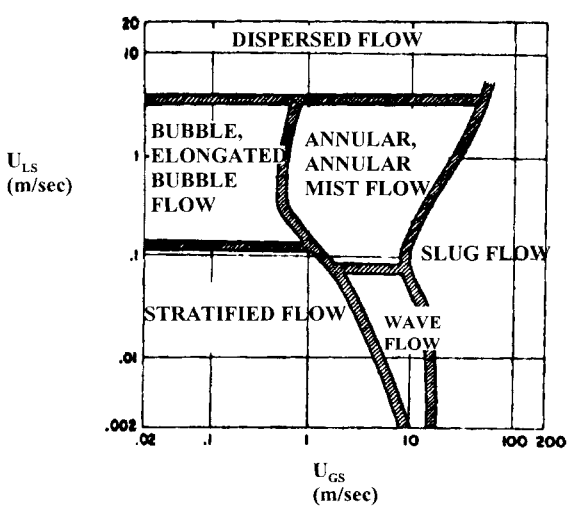

Figure 24. *Example of flow pattern map for air water system in horizontal pipes.*

Maps based on dimensionless coordinates using coordinates such as the Froude number, gas volumetric ratio, Reynolds number, liquid, and gas mass flow rate ratio have been suggested by numerous investigators. Figure 25 shows a map that uses as coordinates $(V_T/\sqrt{gD})^{1/2}$ versus Q_L/Q_G, where Q_L and Q_G are the liquid and gas volumetric flow rates, respectively. V_T is the average total velocity in the pipe, D is the pipe diameter, and g the acceleration of gravity. Other systems employ a mixed coordinate system. For example, the well known Baker plot (Figure 26) uses G/λ versus $L\lambda\psi/G$, where G is the gas mass velocity (a dimensional coordinate) and L/G is a ratio of liquid and gas mass flow rates. The fluid properties enter as the dimensionless correction factors defined by the following expressions:

$$\lambda = [(\rho_G / 0.075)\ (\rho_L / 62.3)]^{1/2}$$

$$\psi = (73.0/\sigma)\ [(\mu_L / 1.0)\ (62.3 / \rho_L)^2\ ]^{1/3}$$

where ρ_G and ρ_L are the gas and liquid densities (in units of lb/ft^3), μ_L is the liquid viscosity (centipoise), and σ is the surface tension (dynes/cm). Flow pattern maps that are based upon dimensionless coordinates are generally thought to be more reliable since they are based upon more universal parameters and are independent of fluid properties and system geometric parameters. However, dimensionless plots are not always easy to construct and the selection of the proper parameters for dimensionless groups is not trivial.

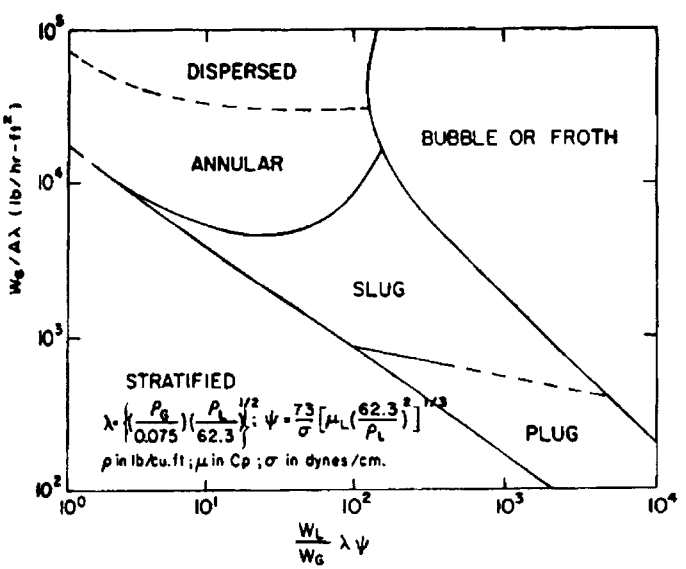

Figure 25. *Baker flow map.*

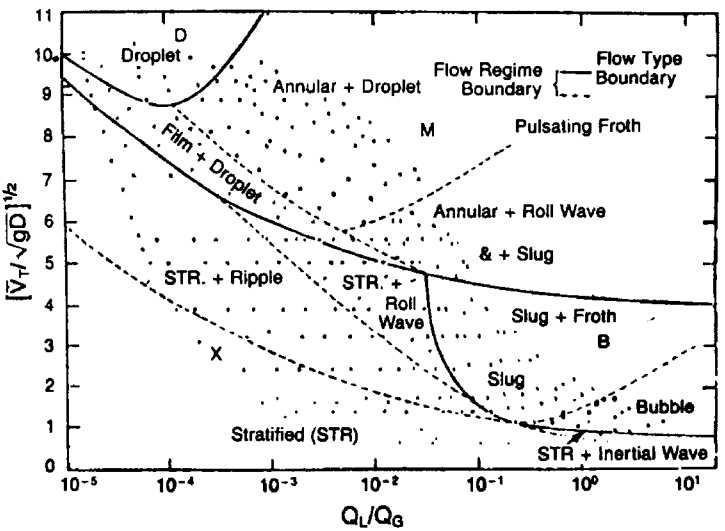

Figure 26. *Flow regime map for horizontal gas and liquid flows (D - droplet; X - stratification; M - mixed; B - bubble).*

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Dimensional analysis is a straightforward approach to select the proper dimensionless coordinates. The approach basically involves guessing what the physical variables that affect the flow patterns are. Under isothermal incompressible conditions these are the liquid and gas flow rates (or the superficial velocities U_{Ls} and U_{Gs}), the liquid and gas densities and viscosities (ρ_L, ρ_G, μ_L and μ_G), pipe diameter (D), pipe roughness (ϵ), pipe inclination angle (β), acceleration of gravity (g), liquid surface tension (σ), and the solid to liquid gas contact wetting angle (γ). The flow pattern transition boundaries should be a function of all these variables. Note that there is already a reduced set of variables as it is assumed that the entrance geometry has no influence on the steady state flow pattern. Using the Buckingham π method one can show that the flow pattern boundaries depend on the following dimensionless parameters:

$$(U_{Ls}\rho_L D)/\mu_L,\ (U_{Gs}\rho_L D)\mu_L,\ g\rho_L{}^2D^3/\mu_L{}^2,\ \rho_G/\rho_L,\ \mu_G/\mu_L,\ \sigma\ \rho_L D/\mu_L{}^2,\ \beta,\ \epsilon/D,\ \gamma$$

Note that this list of dimensionless parameters is by no means unique. A set of variables in which each variable in the new set is a combination of the abovementioned set is also permissible and is in principle completely equivalent to the original set. In fact, an infinite number of sets of the dimensionless parameters exist, each of which could be justified as the "original" set. The normal approach at this point is to find an explicit functional relation among one set of variables

based on experimental data. To find the relation among nine parameters based only upon experimental data is impractical. Furthermore, the rigorous result is not a two-dimensional map but rather a nine-dimensional map. In order to reduce the number of dimensionless groups one tries to distinguish among the important and non-important variables. The correlation of the data is performed with relation to the important variables in the hope that the number of important parameters will be only two or three so that the result could be plotted on a two-dimensional map. Note that the number of important parameters depends also on the choice of the dimensionless set of variables. An alternative set of parameters, for example, is:

$$(U_{Ls} + U_{Gs})/(gD)^{1/2},\ U_{Ls}/U_{Gs},\ g\rho_L^2D^3/\mu_L^2,\ \rho_G/\rho_L,\ \mu_G/\mu_L,\ \sigma\rho_L D/\mu_L^2,\ \beta,\ \epsilon/D,\ \gamma$$

In addition to flow regime, hold-up and pressure drop are two other important parameters in two-phase gas-liquid flows. Hold-up is defined as the relative portion of space occupied by a phase in the pipe. It can be expressed on a time or space average basis, with the actual method chosen depending on the intended use of the hold-up value, and the measurement method employed. There are numerous correlations in the literature for hold-up, but most are based upon a pressure drop-hold-up correlation. The following expression is a widely recognized empirical relationship between hold-up and pressure drop:

$$(dP/dL)_F/(dP/dL)_{SL} = a/(1-R_G)^b = \phi_L^2$$

A plot of $(dP/dL)_F/(dP/dL)_{SL}$ versus $(1-R_G)$ results in three distinct lines shown in Figure 27, where the values of the constants a and b are indicated. The three ranges correspond to approximately the bubble-slug, stratified, and annular flow regimes. Formulas to calculate R_G for substitution into the above equation are:

$$(1-R_G) = [\![4+(8/7)d_1]\!]/[\![5+d_1(\beta/(1-\beta))+8/7]\!]$$

where

$$d_1 = 4[\![0.69 +(1-\beta)(4+21.9(V_{LS}/(gD)^{1/2}]\!]Re_{SL}^{1/8}\ (\rho_G/\rho_L)^{1/2}$$

An extensive treatment of gas-liquid flows encountered in industry applications, along with numerous design correlations can be found in Volume 3 of the Encyclopedia of Fluid Mechanics - Gas-Liquid Flows (N. P. Cheremisinoff, editor, Gulf Publishing Co, Houston, TX, 1986). Further discussions in this volume can be found in Chapter 4 with regard to flow regimes typically encountered in bubble columns and similar devices.

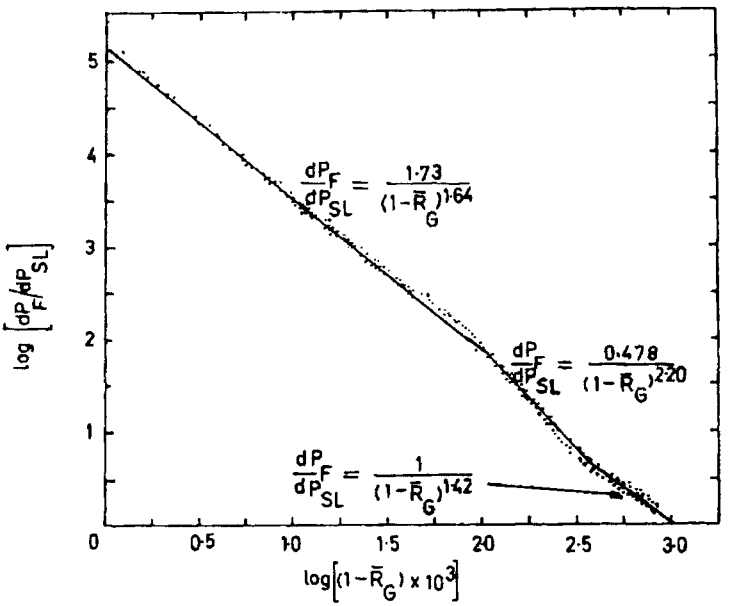

Figure 27. *Empirical correlation between pressure drop and hold-up.*

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DRYING EQUIPMENT

Drying Mechanisms

Drying is a basic unit operation that is applied as a pretreatment step, an intermediate step, or as a finishing step in the preparation of different products. Although drying operations sometimes have a specific objective in a few specialized industries, in general applications throughout the CPI the objective is the removal of water from a system or structure, when the amount-of water present is comparatively small. In the immediate discussions we focus only on the most common applications, whereby the material to be dried may be either a gas, a liquid or a solid. If we are handling a gas, the water may be carried simply as vapor, or in addition it may be present in the form of a spray, fog or mist. If a liquid, water is removed by drying only when present in small amounts; otherwise the process is described as evaporation or distillation. For example - alcohol or acetone may be "dried" by removing the water therefrom. In contrast, for solids the amount of water may vary from a mere trace to the large amount present as in fresh fruit or

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Chapter 4
DISTILLATION EQUIPMENT

INTRODUCTION

Whenever two materials at any temperature possessed a marked difference in their vapor pressures, this difference could be made the basis of a method of separation of the two; and if one or both of the substances were recovered by condensation, the process is called distillation. An equilibrium is established at any temperature between a liquid or a mixture of liquids held in a containing vessel, and the vapor existing above it. Since it is this vapor which ultimately passes out of the containing vessel and when condensed forms the product of the process, the question arises, what is the relation of the composition of this vapor to the composition of the liquid from which it came and with which it is in equilibrium? The answer to this question differs depending on the manufacturing problems of different kinds of mixtures. Obviously, if a volatile component is to be separated from a nonvolatile one, the operation is comparatively simple, and has in principle been described in Chapter 3 under the topic of evaporation. It is necessary only to volatilize the material in one vessel and to condense it in another to accomplish this purpose. In a mechanical mixture of two mutually insoluble liquids sufficiently well agitated to prevent stratification (for example, turpentine and water, heavy fatty acids and water), a dynamic equilibrium is established between the liquid and the vapor when the number of each kind of molecules exiting the liquid is in balance with the number of molecules of each kind again entering the liquid. The pressure existing over the liquid mixture will, therefore, be the sum of these individual pressures, and it follows that a pressure equal to the atmosphere will be reached and boiling will therefore result at a temperature below that at which either of the component liquids would boil alone. This description forms the basis of one type of distillation in which a high-boiling liquid is distilled at a comparatively low temperature by the injection of some low-boiling immiscible liquid or its vapor into the still. In the case where two liquids are mutually soluble, the vapor pressure of each is decreased by the presence of the other, and therefore the sum of their vapor pressures is less than the sum of the vapor pressures of the two liquids before mixing. The composition of the vapor is not independent of the relative amount of the components of the mixture, but is profoundly influenced by each other. This composition can be calculated from the known vapor pressures of the individual pure liquids. When the molecules of two liquids are of relatively the same size, and when there are no

complicated effects when mutually dissolved, such as molecular association, then the composition of the vapor is given by Raoult's law. This law states that that part of the total vapor pressure of a solution of two liquids which is caused by one of the components will equal the product of the vapor pressure of that component in its pure state, and its mole fraction in the liquid. This law, together with Henry's law, are in principle applied in the design of distillation apparatus. In general, the composition of the vapor arising from a solution of one liquid in another is an empirical function of the composition of the solution and must be experimentally determined.

This chapter provides elementary discussions on the subject of distillation, with emphasis given to petroleum refining operations. General terminology and concepts are presented to provide the process engineer with a working knowledge of this important subject, and to become acquainted with the major types of equipment used in distillation operations. Physical properties of hydrocarbon based materials are included in the chapter, as derived from various literature sources and a data base developed by the author over the years as a consultant working in the refining end of the CPI.

OVERVIEW OF DISTILLATION

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Distillation is simply defined as a process in which a liquid or vapor mixture of two or more substances is separated into its component fractions of desired purity, by the application and removal of heat. The process is based on the fact that the vapor of a boiling mixture will be richer in the components that have lower boiling points. Hence, when this vapor is cooled and condensed, the condensate will contain more volatile components. At the same time, the original mixture will contain more of the less volatile material. The primary equipment employed in the process of distillation are distillation columns, which are designed to achieve this separation efficiently. Although the layman has a fair idea as to what "distillation" means, the important aspects that seem to be missed from the manufacturing point of view are: (1) distillation is the most common separation technique; (2) it consumes enormous amounts of energy, both in terms of cooling and heating requirements; (3) it can contribute to more than 50% of plant operating costs. The best way to reduce operating costs of existing units, is to improve their efficiency and operation via process optimization and control. To achieve this improvement, a thorough understanding of distillation principles and how distillation systems are designed is essential.

As stated, distillation is the process of heating a liquid until some of its ingredients pass into the vapor phase, and then cooling the vapor to recover it in liquid form by condensation. The main purpose of distillation is to separate a mixture by taking advantage of different substances' readiness to become a vapor. If the difference in boiling points between two substances is great, complete separation may be easily

accomplished by a single-stage distillation. If the boiling points differ only slightly, many redistillations may be required. In the simplest mixture of two mutually soluble liquids with similar chemical structures, the readiness to vaporize of each is undisturbed by the presence of the other. The boiling point of a 50-50 mixture, for example, would be halfway between the boiling points of the pure substances, and the degree of separation achieved by a single distillation would depend only on each substance's readiness to vaporize at this temperature. This simple law was first stated by 19th- century by the French chemist Frangois Marie Raoult (known as *Raoult's law*).

The term "still" is applied only to the vessel in which liquids are boiled during distillation, but the term is sometimes applied to the entire apparatus, including the fractionating column, the condenser, and the receiver in which the distillate is collected. If a water and alcohol distillate is returned from the condenser and made to drip down through a long column onto a series of plates, and if the vapor, as it rises to the condenser, is made to bubble through this liquid at each plate, the vapor and liquid will interact so that some of the water in the vapor condenses and some of the alcohol in the liquid vaporizes. The interaction at each plate is equivalent to a redistillation. This process is referred to by several names in the industry; namely rectification, fractionation, or fractional distillation.

If two insoluble liquids are heated, each is unaffected by the presence of the other and vaporizes to an extent determined only by its own nature. Such a mixture always boils at a temperature lower than is true for either substance alone. This effect may be applied to substances that would be damaged by overheating if distilled in the usual fashion. Substances can also be distilled at temperatures below their normal boiling points by partially evacuating the still. The greater the vacuum, the lower the distillation temperature.

Basic Components of Distillation Columns

There are a variety of configurations for distillation columns, each designed to perform specific types of separations. A simplified way of classifying distillation columns is to look at how they are operated. In this manner, the two major types are batch and continuous columns. In a batch operation, the feed to the column is introduced batch-wise. That is, the column is charged with a 'batch' and then the distillation process is conducted. When the desired separation is achieved, a next batch of feed is introduced. In contrast, continuous columns process a continuous feed stream. No interruptions occur unless there is a problem or upsets with the column or surrounding process units. They are capable of handling high throughputs and are the most common of the two types. The following discussions focus on continuous columns. Continuous columns can be further classified

according to: (1) the nature of the feed that they are processing (*binary column* - feed contains only two components, and *multi-component column* - feed contains more than two components); (2) the number of product streams they have *(multi-product column* - column has more than two product streams); (3) where the extra feed exits when it is used to help with the separation (*extractive distillation* - where the extra feed appears in the bottom product stream , and *azeotropic distillation* - where the extra feed appears at the top product stream); (4) the type of column internals (*tray column* - where trays of various designs are used to hold up the liquid to provide better contact between vapor and liquid, and hence achieve better separation, and the *packed column* - where instead of trays, *packings* are employed to effect contact between vapor and liquid).

There are several important components in a distillation column, each of which is used either to transfer heat energy or enhance mass transfer. The major components in a typical distillation are:

- a vertical shell where the separation of liquid components is carried out,
- column internals such as trays/plates and/or packings which are used to enhance component separations,
- a reboiler to provide the necessary vaporization for the distillation process,
- a condenser to cool and condense the vapor leaving the top of the column,
- a reflux drum to hold the condensed vapor from the top of the column. The liquid (reflux) is recycled back to the column.

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The column internals are housed within a vertical shell, and together with the condenser and reboiler, constitute a distillation column. A schematic of a typical distillation unit with a single feed and two product streams is shown in Figure 1.

The liquid mixture that is to be processed is called the feed. The feed is introduced usually somewhere near the middle of the column to a tray known as the feed tray. The feed tray divides the column into a top (enriching or rectification) section and a bottom (stripping) section. The feed flows down the column where it is collected at the bottom in the reboiler. Heat is supplied to the reboiler to generate vapor. The source of heat input can be any suitable fluid, although in most chemical plants this is normally steam. In refineries, the heating source may be the output streams of other columns. The vapor raised in the reboiler is re-introduced into the unit at the bottom of the column. The liquid removed from the reboiler is known as the bottoms product or simply, the *bottoms*. Refer to Figure 2 for a simplified view. The vapor travels up the column, and as it exits the top of the unit, it is cooled by a condenser. The condensed liquid is stored in a holding vessel known as the reflux drum. Some of this liquid is recycled back to the top of the column and this is called the reflux. The condensed liquid that is removed from the system is known as the distillate or top product. Refer to Figure 3 for a simplified view.

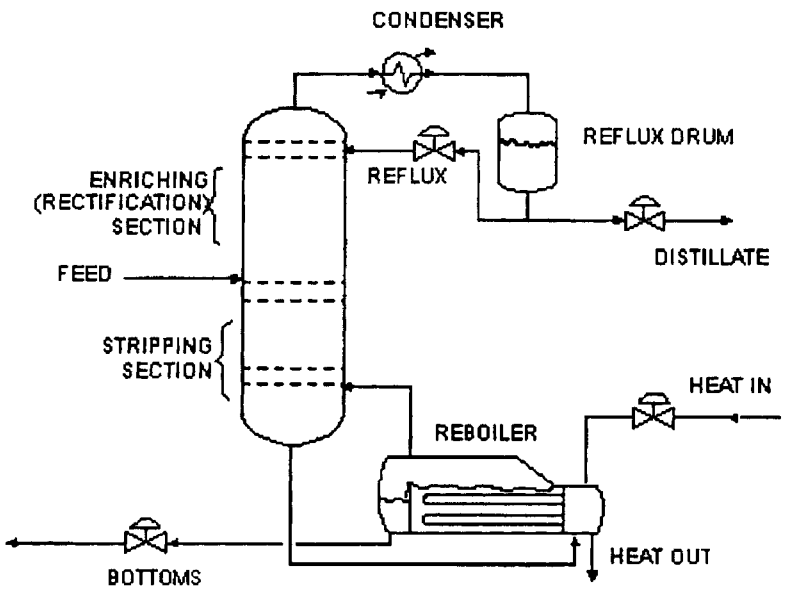

Figure 1. *Basic components of distillation.*

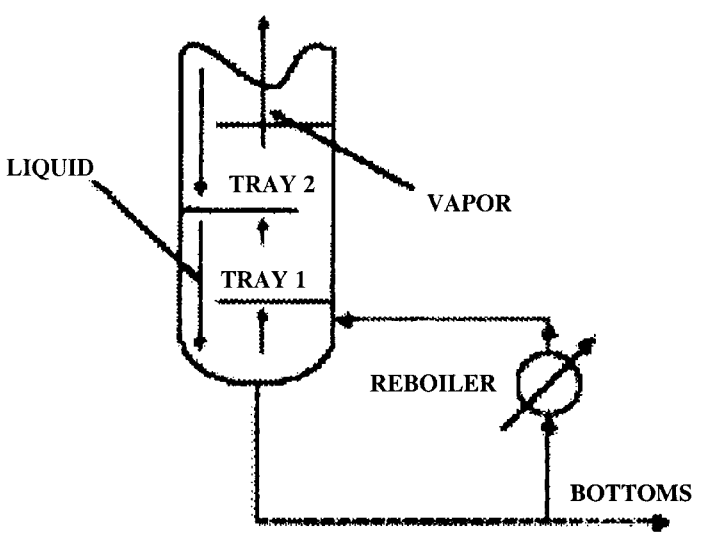

Figure 2. *Diagram shows role of the reboiler.*

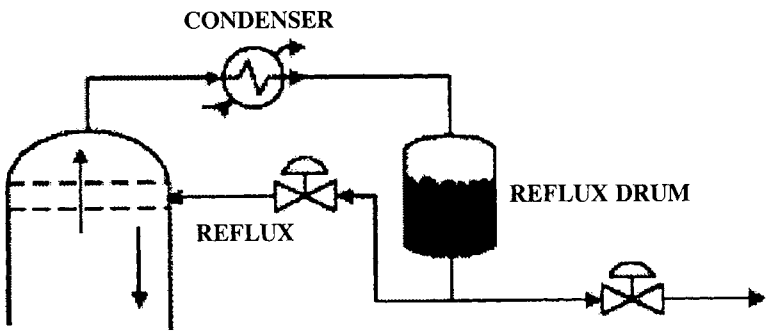

Figure 3. *The role of reflux.*

The terms "trays" and "plates" are used interchangeably. There are many types of tray designs, but the most common ones are bubble cap trays, valve trays, and sieve trays. A bubble cap tray has a riser or chimney fitted over each hole, and a cap that covers the riser. The cap is mounted so that there is a space between riser and cap to allow the passage of vapor. Vapor rises through the chimney and is directed downward by the cap, finally discharging through slots in the cap, and finally bubbling through the liquid on the tray. Bubble-tray or plate towers typically consist of a number of shallow plates or trays over each of which the liquid flows in turn on its way down the tower. The gas enters at the bottom of the tower and is made to flow through a number of bubble caps on each plate. These caps may be of various shapes, though they usually take the form of inverted cups, and their edges are slotted so that the gas escapes from them into the liquid in the form of bubbles. The layout of a typical plate is illustrated Figure 4. The illustration shows the arrangement of bubble caps on a plate together with the risers through which the gas enters the bubble caps and the downcomers which carry the liquid from plate to plate. The inlet weir assists in distributing the liquid over the plate, while the outlet weir maintains the desired depth of liquid. Bubble-plate towers may be preferred to packed towers when: (a) the liquid rate is so low that a packed tower could not be used effectively since the packing would not be adequately wetted; (b) when a difficult distillation duty is required; (c) there is a risk that solid matter may be deposited. Bubble-plate towers, which can be fitted with manholes, are more easily cleaned than packed tower configurations.

In valve trays, perforations are covered by liftable caps. Vapor flows lifts the caps, thus self creating a flow area for the passage of vapor. The lifting cap directs the vapor to flow horizontally into the liquid, thus providing better mixing than is possible in sieve trays.

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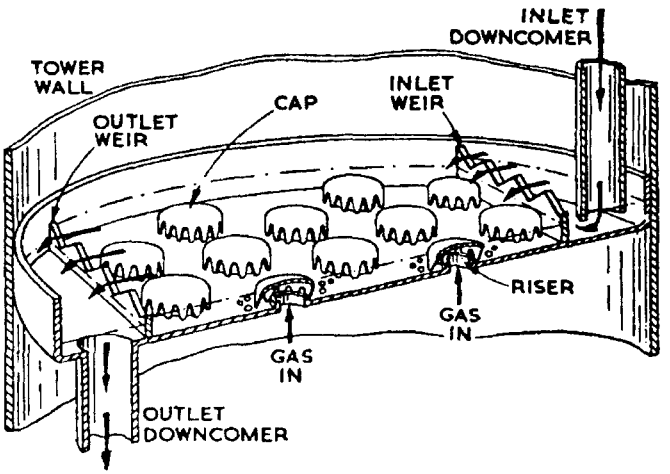

Figure 4. *Bubble cap tray configuration.*

Sieve trays are simply metal plates with holes in them. Vapor passes straight upward through the liquid on the plate. The arrangement, number and size of the holes are design parameters. Because of their efficiency, wide operating range, ease of maintenance and cost factors, sieve and valve trays have replaced the once highly thought of bubble cap trays in many applications.

The flow of liquid and vapor through a tray column is complex. Liquid falls through the downcomers by gravity from one tray to the one below it (refer to Figure 5). A weir on the tray ensures that there is always some liquid (holdup) on the tray and is designed such that the holdup is at a suitable height, e.g., such that the bubble caps are covered by liquid. The vapor flows up the column and is forced to pass through the liquid, via the openings on each tray. The area allowed for the passage of vapor on each tray is called the active tray area. The hotter vapor flows through the liquid on the tray above, and transfers heat to the liquid. During this process some of the vapor condenses adding to the liquid on the tray. The condensate, however, is richer in the less volatile components that is in the vapor. In additionally, because of the heat input from the vapor, the liquid on the tray boils, generating more vapor. This vapor, which moves up to the next tray in the column, is richer in the more volatile components. This continuous and intimate contacting between vapor and liquid occurs on each tray in the column and brings about the separation between low boiling point components and those with higher boiling points. In essence, a tray serves as a mini-column, with each one contributing its share to the overall separation. As such, the more trays there are in a column, the better the degree of separation. Hence, the overall separation efficiency depends significantly on the design of the tray. Trays are designed to maximize vapor-liquid contacting, and hence focus is given to the extent of liquid distribution and vapor distribution achieved by the design.

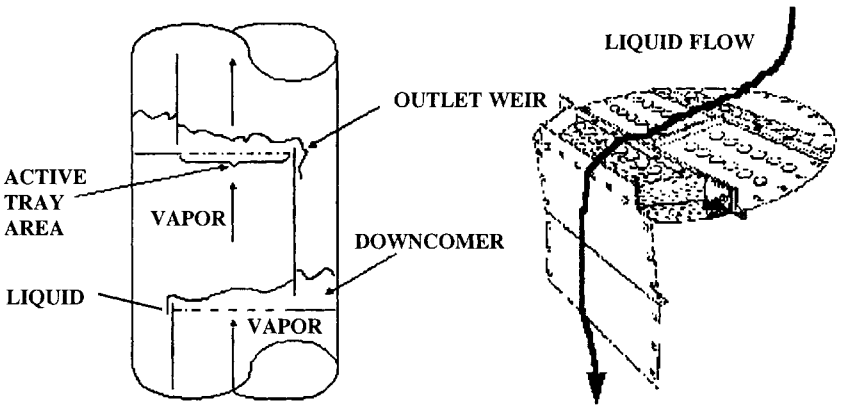

Figure 5. *Downcomer operation and flow over a tray.*

Figure 6. *Sketch of common packing configurations.*

The more intimate the contacting between vapor and liquid, the better the separation that each tray achieves. This means that fewer trays will be needed in order to achieve the same degree of separation. This will result in lower construction costs and energy consumption. Trays alone do not always provide the intimate contact sought. As such, tray designs are sometimes assisted by the addition of packing configurations. Packings are simply passive objects that are designed to increase the interfacial area available for vapor-liquid contacting. Additional discussions on packing materials can be found in Chapter 5, under the section dealing with absorption. Figure 6 illustrates some common geometries of inert packing materials often used in distillation applications. Their role is simply to provide additional surface contact between the vapor and liquid in the column, and to do so without introducing excessive pressure drop across the column. High pressure drop means that more energy is needed to drive the vapor up through a distillation column, and as such there would be higher operating costs. Another very important reason why

inert packing materials are considered is in debottlenecking a column. A tray column that is facing throughput problems can be debottlenecked by replacing a section of trays with packing. The packing will provide additional interfacial contact area for the liquid-vapor contact, thereby increasing the efficiency of the separation for the same column height. In addition, packed columns tend to be shorter than tray-type columns. The packed column is often referred to as a continuous-contact column, whereas a trayed column is called a staged-contact column because of the manner in which the vapor and liquid come into contact.

The function of reboilers has already been discussed. These components are essentially heat exchangers that are used to transfer heat to bring the liquid at the bottom of the column to its boiling point (refer also to discussions in Chapter 1). The principle types employed are jacketted kettles, simple kettle type reboilers, internal reboilers, and thermo-syphon reboilers. Examples of each type are illustrated in Figure 7.

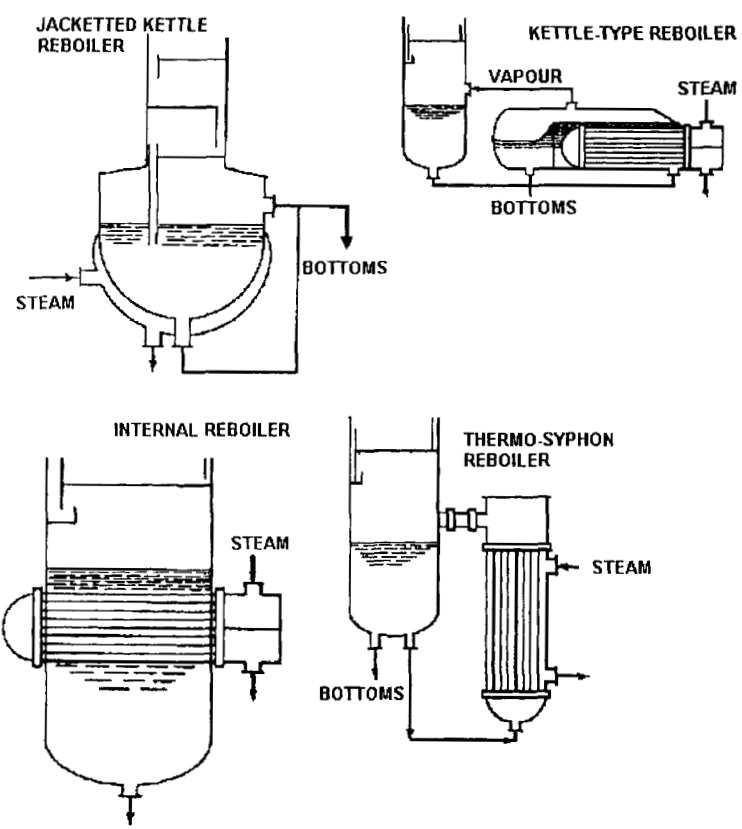

Figure 7. *Common types of reboilers.*

Design Principles

The process of distillation is aimed at the separation of components from a liquid mixture. This process depends on the differences in boiling points of the individual components. Also, depending on the concentrations of the components present, the liquid mixture will have different boiling point characteristics. This means that distillation processes depends on the vapor pressure characteristics of liquid mixtures.

The vapor pressure of a liquid at a particular temperature is the equilibrium pressure exerted by molecules leaving and entering the liquid surface. Some general concepts to recognize about regarding vapor pressure are first, energy input raises the vapor pressure. Also, vapor pressure is related to boiling. A liquid boils when its vapor pressure equals the surrounding pressure. The ease with which a liquid boils depends on its volatility. Liquids with high vapor pressures (i.e., volatile liquids) will boil at lower temperatures. We should also recognize that the vapor pressure and hence the boiling point of a liquid mixture depends on the relative amounts of the components in the mixture. Distillation is accomplished because of the differences in the volatility of the components in a liquid mixture.

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It is the boiling point diagram that provides an understanding of the process. The boiling point diagram shows how the equilibrium compositions of the components in a liquid mixture vary with temperature at a fixed pressure. Consider an example of a liquid mixture containing 2 components: A and B. Figure 8 shows the boiling point diagram for this binary mixture. The boiling point of A is that at which the mole fraction of A is unity. The boiling point of B is that at which the mole fraction of A is zero. In this example, A is the more volatile component and therefore has a lower boiling point than B. The upper curve in the diagram is called the dew-point curve while the lower one is called the bubble-point curve. The region above the dew-point curve shows the equilibrium composition of the superheated vapor while the region below the bubble-point curve shows the equilibrium composition of the subcooled liquid. For example, when a subcooled liquid with mole fraction of A=0.4 (point A) is heated, its concentration remains constant until it reaches the bubble-point (point B), when it starts to boil. The vapors evolved during the boiling has the equilibrium composition given by point C in Figure 8, approximately 0.8 mole fraction A. This is approximately 50% richer in A than the original liquid. This difference between liquid and vapor compositions is the basis for any distillation process.

A term of great importance is relative volatility. Relative volatility is a measure of the differences in volatility between two components, and hence their boiling points. It indicates how easy or difficult a particular separation will be. The relative volatility of component 'i' with, respect to component "j" is defined by the following relationship:

$$\alpha_{ij} = [y_i/x_i] / [y_j/x_j]$$

where y_i is the mole fraction of component "i" in the vapor, and x_i is the mole fraction of component "i" in the liquid. We can conclude that if the relative volatility between two components is very close to one, it is an indication that they have very similar vapor pressure characteristics. This means that they have very similar boiling points and therefore, it will be relatively difficult to separate the two components by means of distillation.

Since the boiling point properties of the components in the mixture being separated are so critical to the distillation process, the vapor-liquid equilibrium (VLE) relationship is of importance. Specifically, it is the VLE data for a mixture which establishes the required height of a column for a desired degree of separation. Constant pressure VLE data is derived from boiling point diagrams, from which a VLE curve can be constructed; like the one illustrated in Figure 9 for a binary mixture. The VLE plot shown expresses the bubble-point and the dew-point of a binary mixture at constant pressure. The curve is called the equilibrium line, and it describes the compositions of the liquid and vapor in equilibrium at a constant pressure condition.

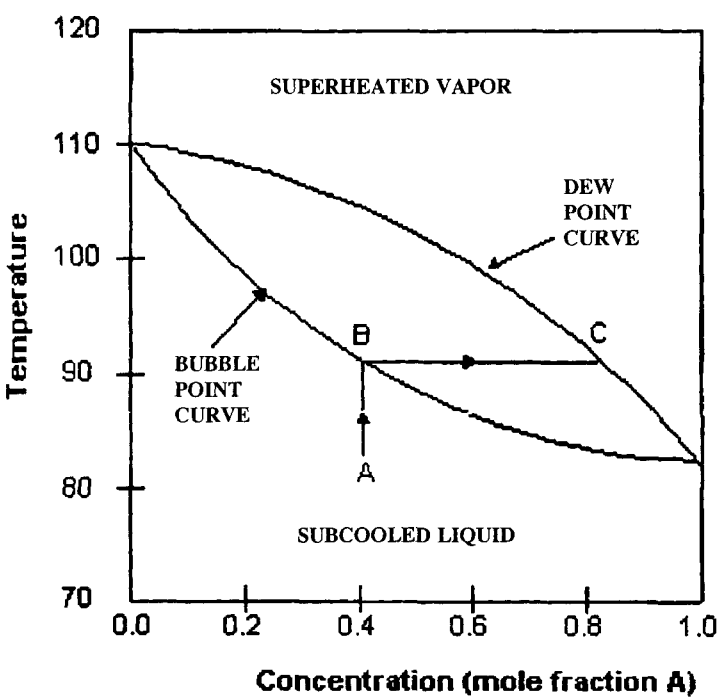

Figure 8. *Boiling point diagram for binary mixture.*

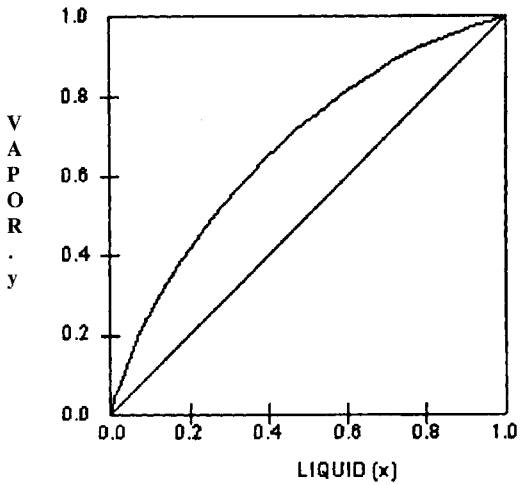

Figure 9. *VLE plot showing equilibrium curve.*

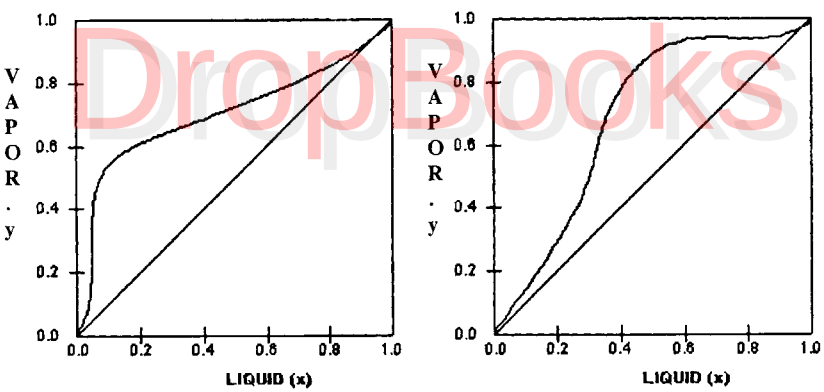

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Figure 10. *Examples of non-ideal VLE curves.*

Figure 9 is the VLE plot for a binary mixture that has essentially a uniform equilibrium, and therefore represents a relatively easy separation. However, there are many cases where non-ideal separations are encountered. These more difficult distillations are defined by the examples shown in Figure 10.

An important system in distillation is an azeotropic mixture. An azeotrope is a liquid mixture which when vaporized, produces the same composition as the liquid. The VLE plots illustrated in Figure 11 show two different azeotropic systems: one with a minimum boiling point and one with a maximum boiling point. In both plots, the equilibrium curves cross the diagonal lines.

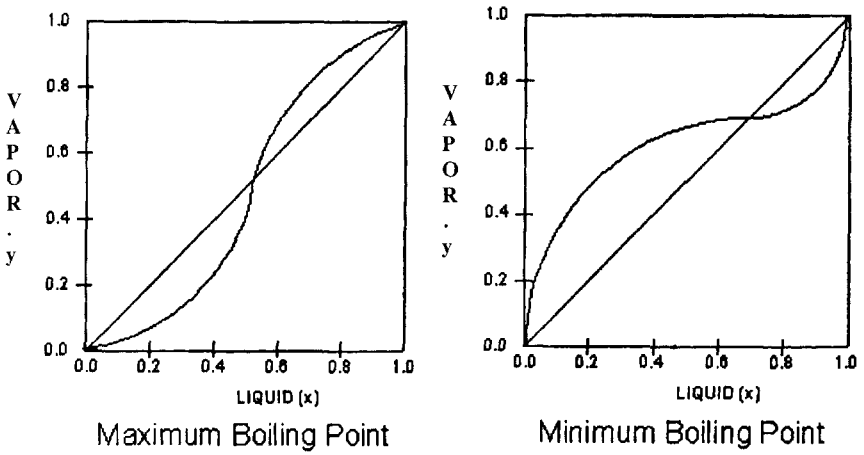

Figure 11. *VLE curves for azeotropic systems.*

These are azeotropic points where the azeotropes occur. In other words, azeotropic systems give rise to VLE plots where the equilibrium curves crosses the diagonals. Both plots are however, obtained from homogenous azeotropic systems. An azeotrope that contains one liquid phase in contact with vapor is called a homogenous azeotrope. A homogenous azeotrope cannot be separated by conventional distillation. However, vacuum distillation may be used as the lower pressures can shift the azeotropic point. Alternatively, an additional substance may added to shift the azeotropic point to a more favorable position. When this additional component appears in appreciable amounts at the top of the column, the operation is referred to as an *azeotropic distillation*. When the additional component appears mostly at the bottom of the column, the operation is called *extractive distillation*.

Another distinction describing a azeotropic system is illustrated in Figure 12. This plot describes the case of a heterogenous azeotrope. Heterogenous azeotropes can be identified by the flat portion on the equilibrium diagram. They may be separated in two distillation columns since these substances usually form two liquid phases with widely differing compositions. The phases may be separated using settling tanks under appropriate conditions.

The design of a distillation column is based on information derived from the VLE diagram describing the mixtures to be separated. The vapor-liquid equilibrium characteristics are indicated by the characteristic shapes of the equilibrium curves. This is what determines the number of stages, and hence the number of trays needed for a separation. Although column designs are often proprietary, the classical method of McCabe-Thiele for binary columns is instructive on the principles of design.

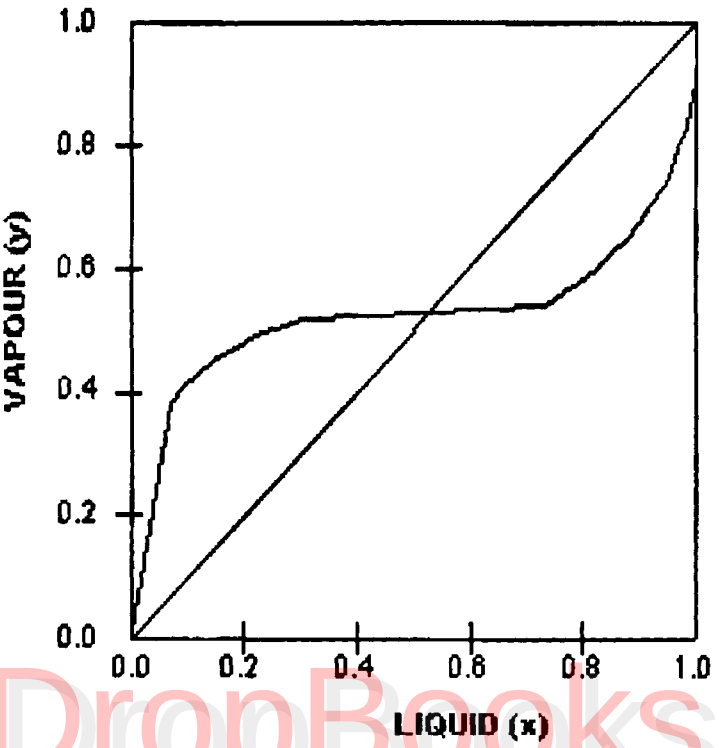

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Figure 12. *VLE curve for a heterogeneous azeotropic system.*

McCabe-Thiele is a graphical design that uses the VLE plot to determine the theoretical number of stages required to effect the separation of a binary mixture. It assumes constant molar overflow. This implies that the molal heats of vaporization of the components are roughly the same. In addition, it is assumed that heat effects (heats of solution, heat losses to and from column, etc.) are negligible, and that for every mole of vapor condensed, 1 mole of liquid is vaporized. The design procedure is as follows. Given the VLE diagram of the binary mixture, operating lines are drawn first. The operating lines define the mass balance relationships between the liquid and vapor phases in the column. There is one operating line for the bottom (stripping) section of the column, and one for the top (rectification or enriching) section of the column. Use of the constant molar overflow assumption also ensures that the operating lines are straight lines. The operating line for the rectification section is constructed as follows: First the desired top product composition is located on the VLE diagram, and a vertical line produced until it intersects the diagonal line that splits the VLE plot in half. A line with slope $R/(R+1)$ is then drawn from this intersection point. Refer to Figure 13 for illustration of the procedure.

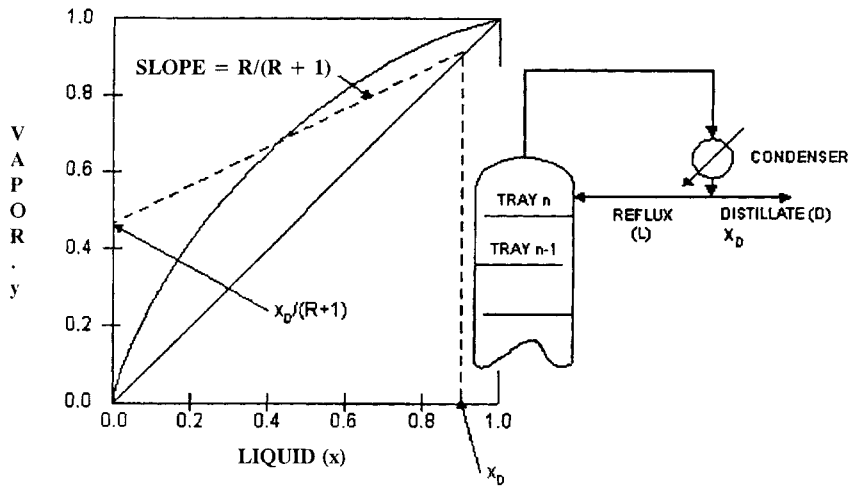

Figure 13. *Application of McCabe-Thiele to VLE diagram.*

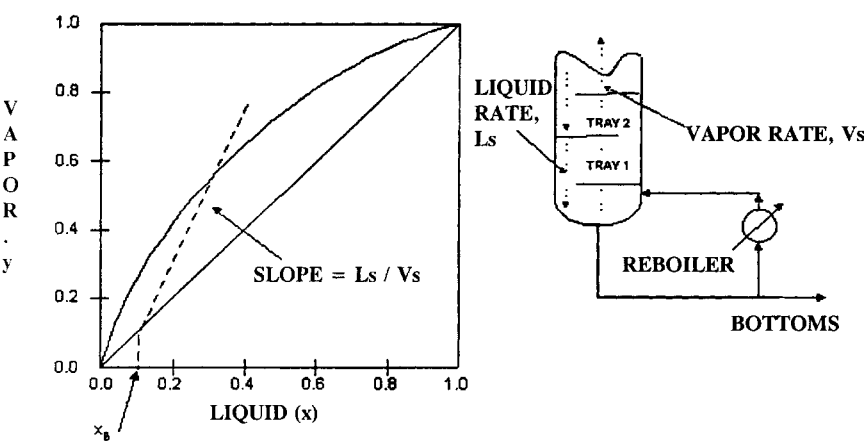

Figure 14. *Construction of operating line for stripping section.*

In Figure 13, R is defined as the ratio of reflux flow (L) to distillate flow (D), and is called the *reflux ratio*. The reflux ratio is a measure of how much of the material is going up through the top of the column and is returned back to the column as reflux. In a similar fashion, the operating line for the stripping section is constructed. In this case the starting point is the desired bottom product composition. A vertical line is drawn from this point to the diagonal line, and a line of slope L_s/V_s is drawn as illustrated in Figure 14. In this figure, L is defined as the liquid rate flowing down the stripping section of the column, while V_s is the vapor

rate of flow up the stripping section of the column. The slope of the operating line for the stripping section is a ratio between the liquid and vapor flows in that part of the column . The McCabe-Thiele method assumes that the liquid on a tray and the vapor above it are in equilibrium. How this is related to the VLE plot and the operating lines is depicted graphically in Figure 15. Figure 15 shows a magnified section of the operating line for the stripping section in relation to the corresponding nth stage in the column. The L's are the liquid flows while the V's represent the vapor flows. Parameters x and y denote liquid and vapor compositions and the subscripts denote the origin of the flows or compositions. The denotation *n - 1* refers to material from the stage below stage n, while *n + 1* refers to material from the stage above stage n. The liquid in stage n and the vapor above it are in equilibrium, therefore, x_n and y_n lie on the equilibrium line. Since the vapor is carried to the tray above without changing composition, this is depicted as a horizontal line on the VLE plot. Its intersection with the operating line provides the composition of the liquid on tray *n + 1*, as the operating line defines the material balance on the trays. The composition of the vapor above the *n +1* tray is obtained from the intersection of the vertical line from this point to the equilibrium line. By repeatedly applying the graphical construction technique, a number of corner sections are created, with each section being equivalent to a stage of the distillation. This is the basis of sizing distillation columns using the McCabe-Thiele graphical design method.

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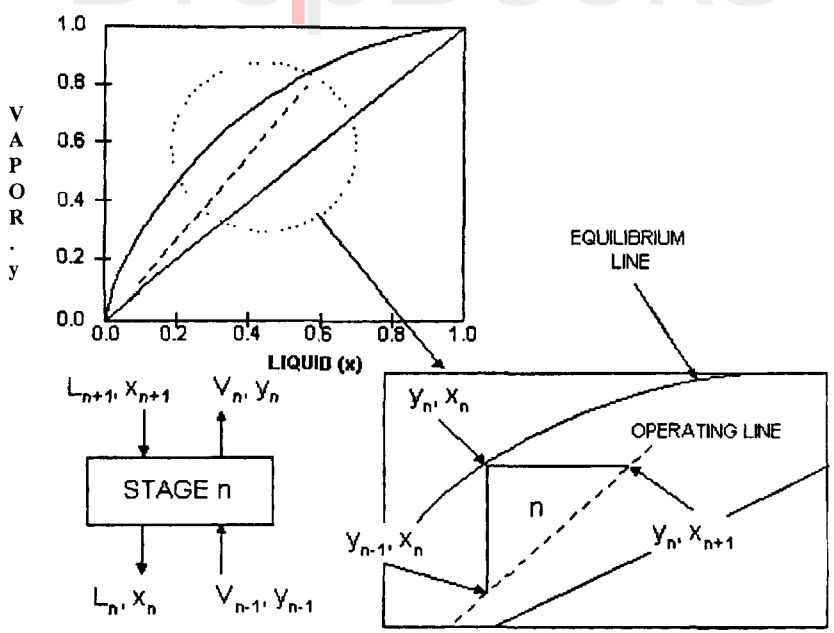

Figure 15. *Illustrates the construction of an operating line.*

From the operating lines for both stripping and rectification sections, the graphical construction described above is further illustrated in Figure 16, which shows that 7 theoretical stages are required to achieve a theoretical separation. The required number of trays (as opposed to stages) is one less than the number of stages since the graphical construction includes the contribution of the reboiler in completing the separation. The actual number of trays required is equal to the ratio of the number of theoretical trays to the tray efficiency. Typical values for tray efficiency ranges from 0.5 to 0.7. Tray efficiency depends on such factors as the type of trays being used, and internal liquid and vapor flow conditions. Sometimes, additional trays are added (up to 10%) to accommodate the possibility that the column may be under-designed. Figure 16 also helps to illustrate that the binary feed should be introduced at the 4th stage. However, if the feed composition is such that it does not coincide with the intersection of the operating lines, this means that the feed is not a saturated liquid. The condition of the feed can be deduced by the slope of the feed line or so-called q-line. The q-line is that line drawn between the intersection of the operating lines, and where the feed composition lies on the diagonal line.

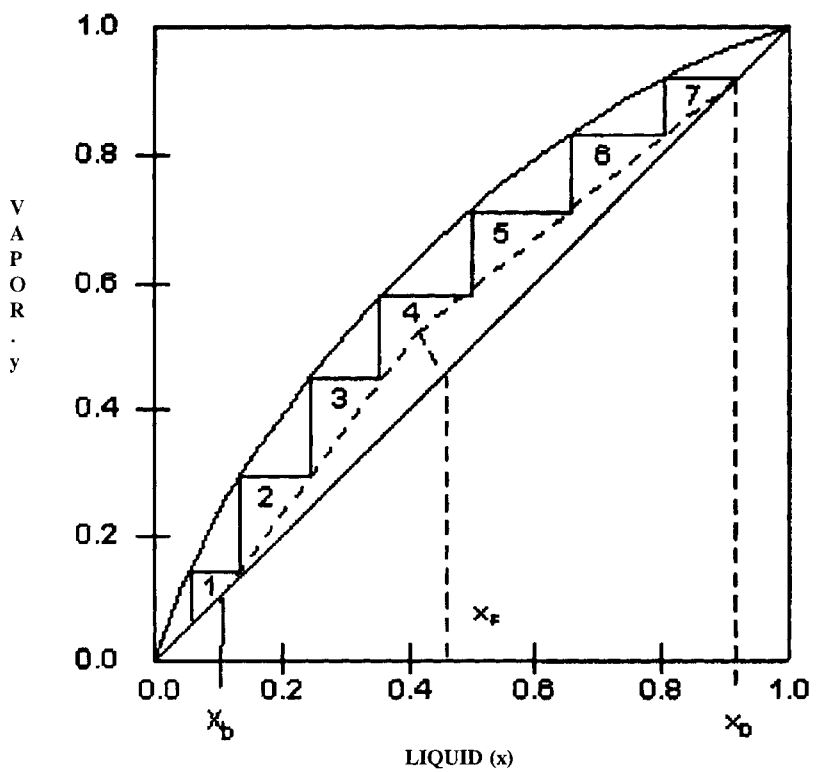

Figure 16. *Example of a seven stage distillation.*

The state of the feed established the slope of the feed line. For examples: saturated vapor exists for $q = 0$; $q = 1$ for saturated liquid; for a mix of liquid and vapor, $0 \leq q \leq 1$; for a subcooled liquid $q \geq 1$; and for a superheated vapor, $q \leq 0$.

From information the feed mixture conditions, the q-line can be constructed and applied in the McCabe-Thiele design. However, excluding the equilibrium line, only two other pairs of lines can be used in the McCabe-Thiele procedure. These are the feed-line and rectification section operating lines, the feed-line and stripping section operating lines, and the stripping and rectification operating lines. The reason for this being that these pairs of lines determine the third.

Determining the number of stages required for the desired degree of separation and the location of the feed tray is only the first step in generating an overall distillation column design. Other factors that need to be considered are tray spacings; column diameter; internal configurations; heating and cooling duties. All of these can lead to conflicting design parameters and trade-offs. Thus, distillation column design is often an iterative procedure. If the conflicts are not resolved at the design stage, then the column will not perform well in practice.

The state of the feed mixture and feed composition not only affects the operating lines and hence the number of stages required for separation, but also the location of the feed tray. During operation, if the deviations from design specifications are excessive, then the column may no longer be able handle the separation task. To overcome the problems associated with the feed, some column are designed to have multiple feed points when the feed is expected to contain varying amounts of components. It is important to note that as the reflux ratio increases, the gradient of operating line for the rectification section moves towards a maximum value of unity. Physically, what this means is that more and more liquid that is rich in the more volatile components are being recycled back into the column. Separation then improves and thus less trays are required to achieve the same degree of separation. Minimum trays are required under total reflux conditions, i.e. there is no withdrawal of distillate. On the other hand, as the reflux decreases, the operating line for the rectification section moves towards the equilibrium line. The 'pinch' between operating and equilibrium lines becomes more pronounced and more and more trays are needed. The McCabe-Thiele method easily verifies this.

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The limiting condition occurs at minimum reflux ration, when an infinite number of trays will be required to effect separation. Most columns are designed to operate between 1.2 to 1.5 times the minimum reflux ratio because this is approximately the region of minimum operating costs (more reflux means higher reboiler duty).

A critical consideration in the design of a distillation column is the vapor flow condition. Improper conditions such as foaming, entrainment, weeping/dumping, and flooding can cause significant inefficiencies in the separation. These conditions are often avoided based upon experienced operating design criteria established. Foaming refers to the expansion of liquid due to passage of vapor, or gas. Although

it provides high interfacial liquid-vapor contact, excessive foaming often leads to liquid buildup on trays. In some cases, foaming may be so excessive that the foam mixes with liquid on the tray above. Whether foaming will occur depends primarily on the physical properties of the liquid mixtures, but is sometimes due to tray designs and conditions. Whatever the cause, separation efficiency is always reduced. Entrainment refers to the liquid carried by vapor up to the tray above and is again caused by high vapor flow rates. It is detrimental because tray efficiency is reduced: lower volatile material is carried to a plate holding liquid of higher volatility. It could also contaminate high purity distillate. Excessive entrainment can lead to flooding. Weeping is a phenomenon caused by low vapor flow. The pressure exerted by the vapor is insufficient to hold up the liquid on the tray. Therefore, liquid starts to leak through perforations on the tray. Excessive weeping will lead to dumping. That is the liquid on all trays will crash (dump) through to the base of the column (via a domino effect) and the column will have to be re-started. Weeping is indicated by a sharp pressure drop in the column and reduced separation efficiency. Flooding occurs due to excessive vapor flow, causing liquid to be entrained in the vapor up the column. The increased pressure from excessive vapor also backs up the liquid in the downcomer, causing an increase in liquid holdup on the plate above. Depending on the degree of flooding, the maximum capacity of the column may be severely reduced. Flooding is detected by sharp increases in column differential pressure and significant decrease in separation efficiency.

Many of the above factors that affect column operation are due to vapor flow conditions being either excessive or too low. Vapor flow velocity is dependent on column diameter. Weeping determines the minimum vapor flow required while flooding determines the maximum vapor flow allowed, hence column capacity. Thus, if the column diameter is not sized properly, the column will not perform well. Not only will operational problems occur, the desired separation duties may not be achieved.

The actual number of trays needed for a particular separation duty depends on the efficiency of the plate, and the packings if they are used. Thus, any factors that cause a decrease in tray efficiency will also change the performance of the column. Tray efficiencies are affected by such factors as fouling, wear and tear and corrosion, and the rates at which these occur depends on the properties of the liquids being processed. Thus the proper materials of construction must be selected for tray construction.

A final consideration is weather conditions. Most distillation columns are open to the atmosphere. Although many of the columns are insulated, varying weather conditions can affect column operation. As such, the reboiler must be appropriately sized to ensure that enough vapor can be generated during cold and windy spells and that it can be turned down sufficiently during hot seasons. The same guideline applies to condensors. Other factors to consider include changing operating conditions and throughputs, brought about by changes in upstream conditions and

changes in the demand for the products. These factors, including the associated control system, should be considered at the design stages because once a column is built and installed, nothing much can be done to rectify the situation without incurring additional significant costs.

With the above as an elementary background to the subject of distillation, we will turn our attention to refinery operations and the equipment typically used. Before doing so, a discussion of the properties of hydrocarbons is provided.

GENERAL PROPERTIES OF HYDROCARBONS

Hydrocarbons are compounds containing only hydrogen and carbon atoms. Since a hydrocarbon is a chemical combination of hydrogen and carbons, both of which are non-metals, hydrocarbons are convalently bonded. Hydrogen has only one electron in the outer ring and, therefore, will form only one bond, by donating one electron to the bond. Carbon, on the other hand, occupies a unique position in the Periodic Table, being halfway to stability with its four electrons in the outer ring. None of these electrons are paired, so carbon uses all of them to form covalent bonds. Carbon's unique structure makes it the basis of organic chemistry.

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Carbon not only combines covalently with other non-metals, but also with itself. Oxygen also reacts with itself to form O_2, hydrogen reacts with itself to form H_2, nitrogen reacts with itself to form N_2, fluorine reacts with itself to form F_2, and chlorine reacts with itself to form Cl_2. Forming diatomic molecules, however, is the extent of the self-reaction of the elemental gases, while carbon has the ability to combine with itself almost indefinitely. Although the elemental gases form molecules when they combine with themselves, the carbon-to-carbon combination must include another element or elements, generally hydrogen. This combination of carbon with itself (plus hydrogen) forms a larger molecule with every carbon atom that is added to the chain. When the chain is strictly carbon-to-carbon with no branching, the resulting hydrocarbon is referred to as a straight-chain hydrocarbon. Where there are carbon atoms joined to carbon atoms to form side branches off the straight chain, the resulting compound is known as a branched hydrocarbon, or an isomer.

The carbon-to-hydrogen bond is always a single bond. While the resulting bond between carbon and hydrogen is always a single bond, carbon does have the capability to form double and triple bonds between itself and other carbon atoms, and/or any other atom that has the ability to form more than one bond. When a hydrocarbon contains only single bonds between carbon atoms, it is known as a saturated hydrocarbon; when there is at least one double or triple bond between two carbon atoms anywhere in the molecule, it is an unsaturated hydrocarbon. When determining the saturation or unsaturation of a hydrocarbon, only the carbon-to-

carbon bonds are considered, since the carbon-to-hydrogen bond is always single. Hydrocarbons are among the most useful materials to mankind, but are also among the most dangerous in terms of their fire potential.

An analogous series of hydrocarbons, and one of the simplest, are the compounds known as the *alkanes*. In this series, the names of all the compounds end in -ane. The first compound in this series is methane. Methane's molecular formula is CH_4. Methane is a gas and is the principal ingredient in the mixture of gases known as natural gas. The next compound is this series is ethane, whose molecular formula is C_2H_6. It is also a gas present in natural gas, although in a much lower percentage than methane. The difference in the molecular formulas of methane and ethane is one carbon and two hydrogen atoms.

Propane is the next hydrocarbon in this series, and its molecular formula is C_3H_8 which is one carbon and two hydrogen atoms different from ethane. Propane is an easily liquified gas which is used as fuel.

The next hydrocarbon in the series is butane, another rather easily liquified gas used as a fuel. Together, butane and propane are known as the LP (liquified petroleum) gases. Butane's molecular formula is C_4H_{10}, which is CH_2 bigger than propane.

Hence, the series begins with a one-carbon-atom compound, methane, and proceeds to add one carbon atom to the chain for each succeeding compound. Since carbon will form four convalent bonds, it must also add two hydrogen atoms to satisfy those two unpaired electrons and allow carbon to satisfy the octet rule, thus achieving eight electrons in the outer ring. In every hydrocarbon, whether saturated or unsaturated, all atoms must reach stability. There are only two elements involved in a hydrocarbon, hydrogen and carbon; hydrogen must have two electrons in the outer ring, and carbon must have eight electrons in the outer ring. Since the carbon-hydrogen bond is always single, the rest of the bonds must be carbon-carbon, and these bonds must be single, double, or triple, depending on the compound.

Continuing in the alkane series (also called the paraffin series because the first solid hydrocarbon in the series is paraffin, or candle wax), the next compound is pentane. This name is derived from the Greek word penta, for five. As its name implies, it has five carbon atoms, and its molecular formula is C_5H_{12}. From pentane on, the Greek prefix for the numbers five, six, seven, eight, nine, ten, and so on are used to name the alkanes, the Greek prefix corresponding to the number of carbon atoms in the molecule. The first four members of the alkane series do not use the Greek prefix method of naming, simply because their common names are so universally accepted: thus the names methane, ethane, propane, and butane.

The next six alkanes are named pentane, hexane, heptane, octane, nonane, and decane. Their molecular formulas are C_5H_{12}, C_6H_{14}, C_7H_{16}, C_8H_{18}, C_9H_{20} and $C_{10}H_{22}$. The alkanes do not stop at the ten-carbon chain however. Since these first ten represent flammable gases and liquids and most of the derivatives of these

compounds comprise the vast majority of hazardous materials encountered, we have no need to go any further in the series. The general formula for the alkanes is C_nH_{2n+2}. The letter n stands for the number of carbon atoms in the molecule. The number of hydrogen atoms then becomes two more than twice the number of carbon atoms. Since there is more than one analogous series of hydrocarbons, one must remember that each series is unique; the alkanes are defined as the analogous series of saturated hydrocarbons with the general formula C_nH_{2n+2}.

Isomers

Within each analogous series of hydrocarbons there exist isomers of the compounds within that series. An isomer is defined as a compound with the same molecular formula as another compound but with a different structural formula. In other words, if there is a different way in which the carbon atoms can align themselves in the molecule, a different compound with different properties will exist.

Beginning with the fourth alkane, butane, we find we can draw a structural formula of a compound with four atoms and ten hydrogen atoms in two ways; the first is as the normal butane exists and the second is as follows, with the name isobutane (refer to Table 1 for properties).

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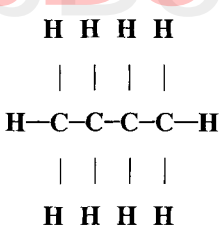
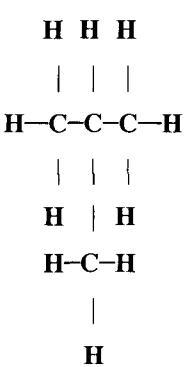
Butane (C_4H_{10}):



Isobutane (C_4H_{10}):



With isobutane, no matter how you count the carbon atoms in the longest chain, you will always end with three. Notice that the structural formula is different - one carbon atom attached to the other carbon atoms - while in butane (also called normal butane), the largest number of carbon atoms another carbon atom can be attached to is two. This fact does make a difference in certain properties of compounds. The molecular formulas of butane and isobutane are the same and, therefore, so are the molecular weights. However, there is a 38-degree difference in melting points, 20-degree difference in boiling points, and the 310-degree difference in ignition temperatures. The structure of the molecule clearly plays part in the properties of the compounds.

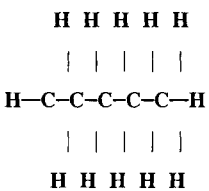
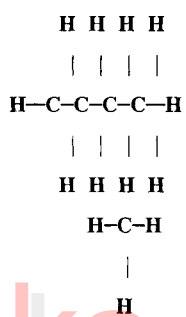
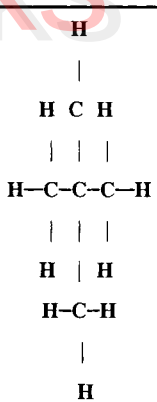
With the five-carbon alkane, pentane, there are three ways to draw the structural formula of this compound with five carbon atoms and twelve hydrogen atoms. The isomers of normal pentane are isopentane and neopentane. The structural formulas of these compounds are illustrated in Table 2, while typical properties are given in Table 1.

Table 1. *Typical Properties of Alkanes*[a]

Compound	Formula	Atomic Weight (°F)	Melting Point (°F)	Boiling Point (°F)	Flash Point (°F)	Ignition Temp. (°F)
Methane	CH_4	16	-296.5	-259	gas	999
Ethane	C_2H_6	30	-298	-127	gas	882
Propane	C_3H_8	44	-306	-44	gas	842
Butane	C_4H_{10}	58	-217	31	gas	550
Pentane	C_5H_{12}	72	-201.5	97	<-40	500
Hexane	C_6H_{14}	86	- 139.5	156	-7	437
Heptane	C_7H_{16}	100	-131.1	209	25	399
Octane	C_8H_{18}	114	-70.2	258	56	403
Nonane	C_9H_{20}	128	-64.5	303	88	401
Decane	$C_{10}H_{22}$	142	-21.5	345	115	410
Butane	C_4H_{10}	58	-217	31	gas	550
Isobutane	C_4H_{10}	58	-255	11	gas	860
Pentane	C_5H_{12}	72	-201.5	97	<-40	500
Isopentane	C_5H_{12}	72	-256	82	<-60	788
Neopentane	C_5H_{12}	72	2	49	<-20	842

(a) - values are average literature reported.

Table 2. *Illustrates the Structural Formulas for Isomers of Pentane.*

COMPOUND	MOLECULAR FORMULA	STRUCTURAL FORMULA
Pentane	C_5H_{12}	
Isopentane	C_5H_{12}	
Neopentane	C_5H_{12}	

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Note the three identical molecular formulas and three identical molecular weights, but significantly different melting, boiling, and flash points and different ignition temperatures. These property differences are referred to as the "structural effect", i.e., differences in the properties of compounds exist for materials having the same molecular formulas but different structural arrangements. This particular structure effect is called the branching effect, and the isomers of all the straight-chain hydrocarbons are called branched hydrocarbons. Another structural effect produced is by the length of the chain formed by consecutively attached carbon atoms.

In noting the increasing length of the carbon chain from methane through decane, the difference in each succeeding alkane is that "unit" made up of one carbon atom and two hydrogen atoms; that "unit" is not a chemical compound itself, but it has a molecular weight of fourteen. Therefore, each succeeding alkane in the analogous series weighs fourteen atomic mass units more than the one before it and fourteen less than the one after it. This weight effect is the reason for the increasing melting and boiling points, the increasing flash points, and the decreasing ignition temperatures. The increasing weights of the compounds also account for the changes from the gaseous state of the first four alkanes, to the liquid state of the next thirteen alkanes, and finally to the solid state of the alkanes, starting with the 17-carbon atom alkane, heptadecane.

Note that the larger a molecule (that is, the greater the molecular weight), the greater affinity each molecule will have for each other molecule, therefore, slowing down the molecular movement. The molecules, duly slowed from their frantic movement as gases, become liquids, and, as the molecules continue to get larger, they are further slowed from their still rapid movement as liquids and become solids.

The straight-chain hydrocarbons represent just one group of straight-chain hydrocarbons, the saturated hydrocarbons known as the alkanes. There are other series of hydrocarbons that are unsaturated; one of those is important in the study of hazardous materials. Additionally, the first hydrocarbon in another series is the only hydrocarbon important in that series. Each of these hydrocarbon series are briefly described below.

Alkenes

The series of unsaturated hydrocarbons that contain just one double bond in the structural formula of each of its members is the analogous series known as the alkenes. Notice that the name of the analogous series is similar to the analogous series of saturated hydrocarbons known as the alkanes, but the structural formula is significantly different. Remembering that the definition of a saturated hydrocarbon is a hydrocarbon with nothing but single bonds in the structural formula and that an unsaturated hydrocarbon is a hydrogen-carbon with at least one multiple bond in the structural formula, then we would expect to find a multiple bond in the structural formulas of the alkenes. The names of all the hydrocarbons in this series end in -ene. The corresponding names for this series of hydrocarbons is similar to the alkanes, with the only difference being the above-mentioned ending. Thus, in the alkene series ethane becomes ethene, propane is propene, butane is butene; the five-carbon straight-chain hydrocarbon in the alkene series is pentene, as opposed to pentane in the alkane series, and so on.

Note that these compounds are covalently bonded compounds containing only hydrogen and carbon. The differences in their structural formulas are apparent; the alkanes have only single bonds in their structural formulas, while the alkenes have one (and only one) double bond in their structural formulas. There are different numbers of hydrogen atoms in the two analogous series. This difference is due to the octet rule that carbon must satisfy. Since one pair of carbon atoms shares a double bond, this fact reduces the number of electrons the carbons need (collectively) by two, so there are two fewer hydrogen atoms in the alkene than in the corresponding alkane.

In any hydrocarbon compound, carbon will form four covalent bonds. In saturated hydrocarbons the four bonds will all be single bonds. The definition of an unsaturated hydrocarbon, however, is a hydrocarbon with at least one multiple bond, and the alkenes are an analogous series of unsaturated hydrocarbons containing just one double bond (which is a multiple bond). The double bond must be formed with another carbon atom since hydrogen atoms can form only single bonds and, in a hydrocarbon compounds there are no other elements but hydrogen and carbon. In forming a double bond with another carbon atom and to satisfy the octet rule, the alkene must form fewer bonds with hydrogen, resulting in less hydrogen in the structural formula of each alkene than in the corresponding alkane.

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There are two fewer hydrogen atoms in each of the alkenes than in the alkane with the same number of carbon atoms. This is also shown by the general molecular formula of the alkenes, C_nH_{2n}, as opposed to the general molecular formula of the alkanes, which is C_nH_{2n+2}.

Note that there is no one-carbon alkene corresponding to methane, since hydrogen can never form more than one covalent bond, and there is no other carbon atom in the structural formula. Therefore, the first compound in the alkene series is ethene, while the corresponding two-carbon compound in the alkane series, ethane, is the second compound in the series, with methane the first.

Although the naming of the alkenes is the same as the alkanes, with only the ending changed from -ane to -ene, there is a problem with the names of the first three alkenes. The systematic names of hydrocarbons came a long while after the simplest (that is, the shortest chain) of the compounds in each series was known and named. In naming the alkanes, the system of using the Greek names for numbers as prefixes begins with pentane, rather than with methane. That situation occurred because methane, ethane, propane, and butane were known and named long before it was known that there was an almost infinite length to the chain that carbon could form and that a systematic naming procedure would be needed. Before the new system was adopted, the common names for the shortest-chain compounds had become so entrenched that those names survived unchanged. Therefore, not only are the first four compounds in the alkane series named differently from the rest of the series, the corresponding two-, three-, and four-carbon compounds are not generally

known as ethene, propene, and butene. Their common names are ethylene, propylene, and butylene.

As noted earlier, more than one compound may have the same molecular formula (isomers), but a structural formula is unique to one compound. In addition, there are many chemicals which possess more than one chemical name, for the same reason mentioned above. The most common organic chemicals are those that have the shortest carbon chains. This fact is also true of their derivatives. The inclusion of a double bond in the structural formula has a profound effect on the properties of a compound. Table 3 illustrates those differences through the properties of alkenes. The presence of a double bond (and, indeed, a triple bond) between two carbon atoms in a hydrocarbon increases the chemical activity of the compound tremendously over its corresponding saturated hydrocarbon. The smaller the molecule (that is, the shorter the chain), the more pronounced this activity is. A case in point is the unsaturated hydrocarbon ethylene. Disregarding the present the differences in combustion properties between it and ethane, ethylene is so chemically active that, under the proper conditions, instead of burning, polymerization, which, if it is uncontrolled, is a much more violent reaction than combustion. This tendency to polymerize is due to the presence of the double bond. The tendency to polymerize decreases as the molecule gets bigger (the chain is longer). Only the first four or five of the straight-chain hydrocarbons are important in the study of hazardous materials. Few, if any, of the isomers of the alkenes are common.

There are other hydrocarbon compounds that contain multiple bonds, however, discussion here is limited to those compounds containing just one multiple bond in their molecules. This is because the compounds containing just one multiple bond are the most valuable commercially and, therefore, the most common. There is, however, a simple way to recognize when you are dealing with a compound that may contain two double bonds; that is a name in which the Greek prefix "di-" is used. As example would be the compound butadiene. Recognize from the first part of the name ("buta-") that there are four carbon atoms in the chain, and that there is a double bond present (the ending "-ene"), however, just before the -ene ending is the prefix "di-," meaning two. Therefore, recognize that you are dealing with a four-carbon hydrocarbon with two double bonds.

As in the alkanes, it is possible for carbon atoms to align themselves in different orders to form isomers. Not only is it possible for the carbon atoms to form branches which produce isomers, but it is also possible for the double bond to be situated between different carbon atoms in different compounds. This different position of the double bond also results in different structural formulas, which, of course, are isomers. Just as in the alkanes, isomers of the alkenes have different properties. The unsaturated hydrocarbons and their derivatives are more active chemically than the saturated hydrocarbons and their derivatives.

Table 3. *Typical Properties of Alkenes*

Compound	Formula	Molecular Weight	Melting Point (°F)	Boiling Point (°F)	Flash Point (°F)	Ignition Temp. (°F)
Ethylene	C_2H_4	28	-272.2	-155.0	gas	1,009
Propylene	C_3H_6	42	-301.4	-53.9	gas	927
1-Butene	C_4H_8	56	-300.0	21.7	gas	700
2-Butene	C_4H_8	56	-218.2	38.7	gas	615
1-Pentene	C_5H_{10}	70	-265.0	86.0	32	523
2-Pentene	C_5H_{10}	70	-292.0	98.6	32	NA
1-Hexene	C_6H_{12}	84	-219.6	146.4	-15	487
2-Hexene	C_6H_{12}	84	-230.8	154.4	-5	473
1-Heptene	C_7H_{14}	98	-119.2	199.9	28	500
1-Octene	C_8H_{16}	112	-152.3	250.3	70	446

NA = Not Applicable

Alkynes

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Another analogous series of unsaturated hydrocarbons that contain just one multiple bond, but, instead of being a double bond, it is a triple bond is the alkynes. The names of all the compounds end in -yne. The only compound in this series that is at all common happens to be an extremely hazardous material. It is a highly unstable (to heat, shock, and pressure), highly flammable gas that is the first compound in the series. This two-carbon unsaturated hydrocarbon with a triple bond between its two carbon atoms is called ethyne, and indeed this is its proper name. It is, however, known by its common name, acetylene.

The -ene ending could be confusing, so one must memorize the fact that acetylene is an alkyne rather than an alkene. Its molecular formula is C_2H_2. The fact that it contains this triple bond makes it extremely active chemically, that is what is meant by its instability to heat, shock, and pressure. It takes energy to start a chemical reaction, and heat, shock, and pressure are forms of energy. The fact that the triple bond contains so much energy tied up in the structure means that it will release this energy, which is the input of some slight amount of external energy. When this input energy strikes the molecule of acetylene, the triple bond breaks, releasing the internal energy of the bonds. This produces either great amounts of heat or an explosion, depending on the way in which the external energy was applied.

There are no other alkynes that are of commercial importance, and so acetylene will be the only member of this series that is considered in fire discussions. There are

other alkynes, however, along with hydrocarbons that might have one double bond and a triple bond present in the molecule.

Straight-Chain Hydrocarbon Nomenclature

The system for naming the straight-chain hydrocarbons is based on an agreed-upon method of retaining the first three or four common names, then using Greek prefixes that indicate the number of carbon atoms in the chain. For isomers, the same system is used, always using the name of the compound that is attached to the chain and the name of the chain.

Recall the first analogous series of hydrocarbons the alkanes, a series of saturated hydrocarbons, all ending in -ane. For these hydrocarbons and other hydrocarbons to react, a place on the hydrocarbon chain must exist for the reaction to take place. Since all the bonds from carbon to hydrogen are already used, an "opening" on one of the carbon atoms must exist for it to be able to react with something else. This "opening" occurs when one of the hydrogen atoms is removed from its bond with a carbon atom, thus causing that carbon to revert back to a condition of instability, with seven electrons in its outer ring, or, as we now state, with one unpaired electron. This one unpaired electron (or half of a covalent bond, or "dangling" bond) wants to react with something, and it will, as soon as another particle which is ready to react is brought near. This chain of carbon atoms (from one carbon to another to another, and so on) with a hydrogen atom missing is a particle that was once a compound, and its name is a *radical.*

Radicals are created by energy being applied to them in a chemical reaction or in a fire. Remember that a hydrocarbon compound with at least one hydrogen atom removed is no longer a compound, but a chemical particle known as a radical. Radicals have names of their own; they are derived from the name of the alkane. When a hydrogen atom is removed from the alkane hydrocarbon, the name is changed from -ane to -yl. Therefore, when a hydrogen is removed from the compound methane, the methyl radical is formed. When a hydrogen atom is removed from the compound ethane, the ethyl radical is formed. In the same manner, the propyl radical comes from propane, the butyl radical comes from butane, and so on. Similarily, isobutane will produce the isobutyl radical, and isopentane will produce the isopentyl radical. A list of hydrocarbons and the radicals produced from them when a hydrogen is removed is shown is Table 4. Note that there are only a few radicals from compounds other than the alkanes which are important. Radicals are referred to as hydrocarbon "backbones". As an example, isobutane is more properly named methyl propane. Another isomer with a different proper name is isopentane, more properly called methyl butane. Neopentane is also named 2,2-dimethyl propane.

Table 4. *A Listing of Common Radicals*

Methane	CH_4	Methyl	$-CH_3$
Ethane	C_2H_6	Ethyl	$-C_2H_5$
Propane	C_3H_8	n-Propyl	$-C_3H_7$
		Isopropyl	$-C_3H_7$
Butane	C_4H_{10}	n-Butyl	$-C_4H_9$
Isobutane	C_4H_{10}	Isobutyl	$-C_4H_9$
		sec-Butyl	$-C_4H_9$
		tert-Butyl	$-C_4H_9$
Ethylene	C_2H_4	Vinyl	$-C_2H_3$
Benzene	C_6H_6	Phenyl	$-C_6H_5$

The following is a list of rules for proper nomenclature of the isomers and their derivatives.

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1. Find the longest continuous chain and name it as if it were an alkane.
2. Name the side branches in the same manner.
3. Identify the number of the carbon atom on the longest chain to which the branch is attached by counting from the end of the chain nearest to the branch.
4. If there could is any confusion as to which carbon atom is meant, put the number in front of the name of the compound, followed by a dash.
5. If there is more than one branch, use the numbers to identify the carbon atom to which they are attached.
6. If the branches are identical, use the prefixes di- for two, tri- for three, tetra- for four, and so on.

In this manner, the four isomers of hexane are named 2-methyl pentane, 3-methyl pentane, 2,2,-dimethyl butane, and 2,3-dimethyl butane.

Aromatic Hydrocarbons

The above discussions have concentrated on hydrocarbons, both saturated and unsaturated, with the unsaturated hydrocarbons containing only one multiple bond. The unsaturated hydrocarbons are the alkenes with one double bond and the alkynes

with one triple bond. There are other straight-chain hydrocarbons that are unsaturated containing more than one multiple bond, some with more than one double bond, and some with a mixture of double bonds and triple bonds. The combinations and permutations are endless, but there are only a few of the highly unstable materials.

From a commercial standpoint, there is a large body of hydrocarbons that is very important. These hydrocarbons are different in that they are not straight-chain hydrocarbons but have a structural formula that can only be called cyclical. The most common and most important hydrocarbon in this group is benzene. It is the first and simplest of the six-carbon cyclical hydrocarbons referred to as aromatic hydrocarbons.

Benzene's molecular formula is C_6H_6, but it does not behave like hexane, hexene, or any of their isomers. One would expect it to be similar to these other six-carbon hydrocarbons in its properties. Table 5 provides a comparison between benzene, hexane and 1-hexene. The table shows that there are major differences between benzene and the straight-chain hydrocarbons of the same carbon content. Hexene's ignition temperature is very near to hexane's. The flash point difference is not great, however, there are significant differences in melting points. The explanation for these differences is structure; which in the case of benzene is a cyclical form with alternating double bonds.

The alternating double bonds are illustrated in Figure 17A. Initially, it was believed that the alternating double bonds impart very different properties to benzene, however, and the fact is that they do not. The only possible way for the benzene molecule to exist is illustrated in Figure 17B, in which a circle is drawn within the hexagonal structural to show that the electrons that should form a series of alternating double bonds are really spread among all six carbon atoms. It is the only structure possible that would explain the unique properties of benzene. This structural formula suggests resonance; that is, the possibility that the electrons represented by the circle are alternating back and forth between and among the six carbon atoms.

Table 5. *Compares Between Benzene and of Straight-Chain Hydrocarbons.*

Compound	Formula	Melting Point (°F)	Boiling Point (°F)	Flash Point (°F)	Ignition Temp. (°F)	Molecular weight
Hexane	C_6H_{14}	-139.5	156.0	-7	500	86
1-Hexene	C_6H_{12}	-219.6	146.4	<-20	487	84
Benzene	C_6H_6	41.9	176.2	12	1,044	78

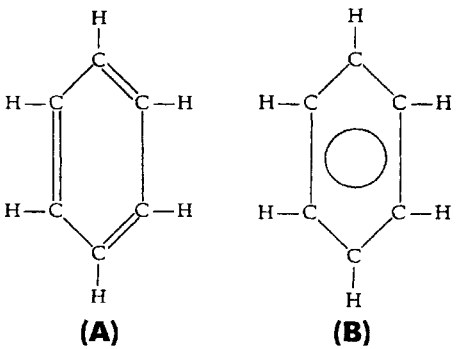

Figure 17. *The structure of benzene: (A) conventional illustration of double bonds, (B) illustration implying resonance.*

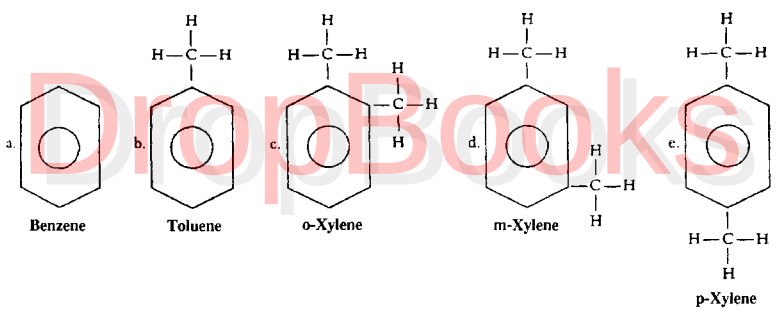

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Figure 18. *The structures of benzene and some of its common derivatives.*

This particular hexagonal structure is found throughout nature in many forms, almost always in a more complicated way, usually connected to many other "benzene rings" to form many exotic compounds. Of importance to the immediate discussions are benzene and a few of its derivatives. Benzene's derivatives include toluene and xylene, whose structural formulas are illustrated in Figure 18 along with that of benzene. Some typical properties are given in Table 6, which illustrates the differences caused by molecular weight and structural formulas. There are other cyclical hydrocarbons, but they do not have the structural formulas of the aromatics, unless they are benzene-based. These cyclical hydrocarbons may have three, four, five, or seven carbons in the cyclical structure, in addition to the six-carbon ring of the aromatics. None of them has the stability or the chemical properties of the aromatics.

Table 6. *Comparison of Benzene and Some of its Derivatives,*

Compound	Formula	Melting Point (°F)	Boiling Point (°F)	Flash Point (°F)	Ignition Temperature (°F)	Molecular Weight
Benzene	C_6H_6	41.9	176.2	12	1,044	78
Toluene	C_7H_8	-138.1	231.3	40	997	92
o-xylene	C_8H_{10}	-13.0	291.2	90	867	106
m-xylene	C_8H_{10}	-53.3	281.9	81	982	106
p-xylene	C_8H_{10}	-55.8	281.3	81	984	106

The aromatic hydrocarbons are used mainly as solvents and as feedstock chemicals for chemical processes that produce other valuable chemicals. With regard to cyclical hydrocarbons, the aromatic hydrocarbons are the only compounds discussed. These compounds all have the six-carbon benzene ring as a base, but there are also three-, four-, five-, and seven-carbon rings. These materials will be considered as we examine their occurrence as hazardous materials. After the alkanes, the aromatics are the next most common chemicals shipped and used in commerce. The short-chain olefins (alkenes) such as ethylene and propylene may be shipped in larger quantities because of their use as monomers, but for sheer numbers of different compounds, the aromatics will surpass even the alkanes in number, although not in volume.

Hydrocarbon Derivatives

A hydrocarbon derivative is a compound with a hydrocarbon backbone and a functional group attached to it chemically. A hydrocarbon backbone is defined as a molecular fragment that began as a hydrocarbon compound and has had at least one hydrogen atom removed from the molecule. Such a fragment is also known as a radical. A functional group is defined as an atom or a group of atoms, bound together, which impart specific chemical properties to a molecule; also referred to as radicals. A hydrocarbon derivative then is essentially a compound made up of two specific parts; the first part comes from a hydrocarbon, and the second may have many different origins (which includes coming from a hydrocarbon), depending on the chemical makeup of the functional group. The hydrocarbon backbone may come from an alkane, an alkene, an alkyne (indeed, any saturated or unsaturated hydrocarbon), or from an aromatic hydrocarbon or other cyclical hydrocarbon. Any hydrocarbon compound may form the hydrocarbon backbone

portion of the hydrocarbon derivative, as long as it has been converted to a radical, by removal of one or more hydrogens, in preparation for the reaction. The functional group may have many origins, with chemists using as reactants any chemical compound that will produce the desired functional group. The functional groups include the halogens (fluorine, chlorine, bromine, and iodine), the hydroxyl radical, the carbonyl group, oxygen, the carboxyl group, the peroxide radical, the amine radical, and even other hydrocarbon radicals. When these functional groups are chemically attached to hydrocarbon backbones, they form compounds called hydrocarbon derivatives, and each functional group imparts a separate set of chemical and physical properties to the molecule formed by this chemical attachment.

Just as the alkanes and alkenes had general formulas, the carbon derivatives all have general formulas. The hydrocarbon backbone provides a portion of the general formula, and the functional group provides the other part. In each case, the hydrocarbon derivative is represented by the formula R-, and the hydrocarbon backbone has its own specific formula. The term "substituted hydrocarbon" is another name for hydrocarbon derivative, because the functional group is substituted for one or more hydrogen atoms in the chemical reaction.

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Halogenated Hydrocarbons

A halogenated hydrocarbon is defined as a derivative of a hydrocarbon in which a hydrogen atom is replaced by a halogen atom. Since all of the halogens react similarly, and the number of hydrocarbons (including all saturated hydrocarbons, unsaturated hydrocarbons, aromatic hydrocarbons, other cyclical hydrocarbons, and all the isomers of these hydrocarbons) is large, the number of halogenated hydrocarbons can also be very large. The most common hydrocarbon derivatives are those of the first four alkanes and the first three alkenes (and, of course, the isomers of these hydrocarbons). There are some aromatic hydrocarbon derivatives, but, again, they are of the simplest structure. Whatever the hydrocarbon backbone is, it is represented in the general formula by its formula, which is R-. Therefore, the halogenated hydrocarbons will have formulas such as R-F, R-Cl, R-Br, and R-I for the respective substitution of fluorine, chlorine, bromine, and iodine on to the hydrocarbon backbone. As a rule, the general formula can be written R-X, with the R as the hydrocarbon backbone, the X standing for the halide (any of the halogens), and the ''-" the covalent bond between the hydrocarbon backbone and the halogen. R-X is read as "alkyl halide".

Radicals of the alkanes are referred to as alkyl radicals. There are two other important radicals; they are the vinyl radical, which is produced when a hydrogen atom is removed from ethylene, and the phenyl radical, which results when a

hydrogen atom is removed from benzene. The term halogenated means that a halogen atom has been substituted for a hydrogen atom in a hydrocarbon molecule. The most common halogenated hydrocarbons are the chlorinated hydrocarbons. The simplest chlorinated hydrocarbon is methyl chloride, whose molecular formula is CH_3Cl. The structural formula for methyl chloride shows that one chlorine atom is substituted for one hydrogen atom. Methyl chloride has many uses, such as a herbicide, as a topical anesthetic, extractant, and low-temperature solvent, and as a catalyst carrier in low-temperature polymerization. It is a colorless gas that is easily liquified and is flammable; it is also toxic in high concentrations. Methyl chloride is the common name for this compound, while its proper name is chloromethane. Proper names are determined by the longest carbon chain in the molecule, and the corresponding hydrocarbon's name is used as the last name of the compound. Any substituted groups are named first, and a number is used to designate the carbon atom that the functional group is attached to, if applicable.

It is possible to substitute more than one chlorine atom for a hydrogen atom on a hydrocarbon molecule; such substitution is done only when the resulting compound is commercially valuable or is valuable in another chemical process. An example is methylene chloride (the common name for dichloromethane), which is made by substituting two chlorine atoms for two hydrogen atoms on the methane molecule. Its molecular formula is CH_2Cl_2. Methylene chloride is a colorless, volatile liquid with a sharp, ether-like odor. It is listed as a non-flammable liquid, but it will ignite at 1,224°F; it is narcotic at high concentrations. It is most commonly used as a stripper of paints and other finishes. It is also a good degreaser and solvent extractor and is used in some plastics processing applications.

Substituting a third chlorine on the methane molecule results in the compound whose proper name is trichloromethane (tri- for three; chloro- for chlorine; and methane, the hydrocarbon's name for the one-carbon chain). It is more commonly known as chloroform. Its molecular formula is $CHCl_3$. Chloroform is a heavy, colorless, volatile liquid with a sweet taste and characteristic odor. It is classified as non-flammable, but it will burn if exposed to high temperatures for long periods of time. It is narcotic by inhalation and toxic in high concentrations. It is an insecticide and a fumigant and is very useful in the manufacture of refrigerants. The total chlorination of methane results in a compound whose proper name is tetrachloromethane (tetra- for four), but its common name is carbon tetrachloride (or carbon tet). This is a fire-extinguishing agent that is no longer used since it has been classified as a carcinogen. It is still present, though, and its uses include refrigerants, metal degreasing, and chlorination of organic compounds. Its molecular formula is CCl_4. It is possible to form analogues of methyl chloride, (methyl fluoride, methyl bromide, methyl iodide), methylene chloride (substitute fluoride, bromide, and iodide in this name also), chloroform (fluoroform, bromoform, and iodoform), and carbon tetrachloride (tetrafluoride, tetrabromide, and tetraiodide). Each of these halogenated hydrocarbons has commercial value.

What was true for one hydrocarbon compound is true for most hydrocarbon compounds, particularly straight-chain hydrocarbons; that is, you may substitute a functional group at each of the bonds where a hydrogen atom is now connected to the carbon atom. Where four hydrogen atoms exist in methane, there are six hydrogen atoms in ethane; you recall that the difference in make-up from one compound to the next in an analogous series is the "unit" made up of one carbon and two hydrogens. Therefore, it is possible to substitute six functional groups on to the ethane molecule. You should also be aware that the functional groups that would be substituted for the hydrogens need not be the same, that is, you may substitute chlorine at one bond, fluorine at another, the hydroxyl radical at a third, an amine radical at a fourth, and so on. Substituting one chlorine atom for a hydrogen atom in ethane produces ethyl chloride, a colorless, easily liquifiable gas with an ether-like odor and a burning taste, which is highly flammable and moderately toxic in high concentrations. It is used to make tetraethyl lead and other organic chemicals. Ethyl chloride is an excellent solvent and analytical reagent, as well as an anesthetic. Its molecular formula is C_2H_5Cl. Although we are using chlorine as the functional group, it may be any of the other halogens. In addition, we are giving the common names, while the proper names may be used on the labels and shipping papers. Ethyl chloride's proper name is chloroethane.

Substituting another chlorine produces ethylene dichloride (proper name 1,2-dichloroethane). In this case, an isomer is possible, which would be the chlorinated hydrocarbon where both chlorines attached themselves to the same carbon atom, whereby 1,1-dichloroethane is formed. These compounds have slightly different properties and different demands in the marketplace. As further chlorination of ethane occurs, we would have to use the proper name to designate which compound is being made. One of the analogues of ethylene dichloride is ethylene dibromide, a toxic material that is most efficient and popular as a grain fumigant, but it is known to be a carcinogen in test animals. There are many uses for the halogenated hydrocarbons. Many of them are flammable; most are combustible. Some halogenated hydrocarbons are classified as neither, and a few are excellent fire-extinguishing agents (the Halons ®), but they will all decompose into smaller, more harmful molecular fragments when exposed to high temperatures for long periods of time.

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Alcohols

The compounds formed when a hydroxyl group (-OH) is substituted for a hydrogen are called alcohols. They have the general formula R-OH. The hydroxyl radical looks exactly like the hydroxide ion, but it is not an ion. Where the hydroxide ion fits the definition of a complex ion - a chemical combination of two or more atoms that have collectively lost or (as in this case) gained one or more electrons - the

hydroxide radical is a molecular fragment produced by separating the -OH from another compound, and it has no electrical charge. It does have an unpaired electron waiting to pair up with another particle having its own unpaired electron. The alcohols, as a group, are flammable liquids in the short-chain range, combustible liquids as the chain grows longer, and finally solids that will burn if exposed to high temperatures, as the chain continues to become longer. As in the case of the halogenated hydrocarbons, the most useful alcohol compounds are of the short-carbon-chain variety. Just as in the case of the halogenated hydrocarbons, the simplest alcohol is made from the simplest hydrocarbon, methane. Its name is methyl alcohol and its molecular formula is CH_3OH.

Nature produces a tremendous amount of methyl alcohol, simply by the fermentation of wood, grass, and other materials made to some degree of cellulose. In fact, methyl alcohol is known as wood alcohol, along with names such as wood spirits and methanol (its proper name; the proper names of all alcohols end in -ol). Methyl alcohol is a colorless liquid with a characteristic alcohol odor. It has a flash point of 54°F, and is highly toxic. It has too many commercial uses to list here, but among them are as a denaturant for ethyl alcohol (the addition of the toxic chemical methyl alcohol to ethyl alcohol in order to form denatured alcohol), antifreezes, gasoline additives, and solvents. No further substitution of hydroxyl radicals is performed on methyl alcohol.

The most widely known alcohol is ethyl alcohol, simply because it is the alcohol in alcoholic drinks. It is also known as grain alcohol, or by its proper name, ethanol. Ethyl alcohol is a colorless, volatile liquid with a characteristic odor and a pungent taste. It has a flash point of 55°F, is classified as a depressant drug, and is toxic when ingested in large quantities. Its molecular formula is C_2H_5OH. In addition to its presence in alcoholic beverages, ethyl alcohol has many industrial and medical uses, such as a solvent in many manufacturing processes, as antifreeze, antiseptics, and cosmetics.

The substitution of one hydroxyl radical for a hydrogen atom in propane produces propyl alcohol, or propanol, which has several uses. Its molecular formula is C_3H_7OH. Propyl alcohol has a flash point of 77°F and, like all the alcohols, burns with a pale blue flame. More commonly known is the isomer of propyl alcohol, isopropyl alcohol. Since it is an isomer, it has the same molecular formula as propyl alcohol but a different structural formula. Isopropyl alcohol has a flash point of 53°F. Its ignition temperature is 850°F, while propyl alcohol's ignition temperature is 700°F, another effect of the different structure. Isopropyl alcohol, or 2-propanol (its proper name) is used in the manufacture of many different chemicals, but is best known as rubbing alcohol.

The above-mentioned alcohols are by far the most common. Butyl alcohol is not as commonly used as the first four in the series, but it is used. Secondary butyl alcohol and tertiary butyl alcohol, so named because of the type of carbon atom in the

molecule to which the hydroxyl radical is attached, must be mentioned because they are flammable liquids, while isobutyl alcohol has a flash point of 100 °F. All of the alcohols of the first four carbon atoms in the alkanes, therefore, are extremely hazardous because of their combustion characteristics.

Whenever a hydrocarbon backbone has two hydroxyl radicals attached to it, it becomes a special type of alcohol known as a glycol. The simplest of the glycols, and the most important, is ethylene glycol, whose molecular formula $C_2H_4(OH)_2$. The molecular formula can also be written CH_2OHCH_2OH and may be printed as such on some labels. Ethylene glycol is a colorless, thick liquid with a sweet taste, is toxic by ingestion and by inhalation, and among its many uses is a permanent antifreeze and coolant for automobiles. It is a combustible liquid with a flash point of 240°F.

The only other glycol that is fairly common is propylene glycol which has a molecular formula of $C_3H_6(OH)_2$. It is a combustible liquid with a flash point of 210°F, and its major use is in organic synthesis, particularly of polyester resins and cellophane.

The last group of substituted hydrocarbons produced by adding hydroxyl radicals to the hydrocarbon backbone are the compounds made when three hydroxyl radicals are substituted; these are known as glycerols. The name of the simplest of this type of compound is just glycerol. Its molecular formula is $C_3H_5(OH)_3$. Glycerol is a colorless, thick, syrupy liquid with a sweet taste, and has a flash point of 320°F, and is used to make such diverse products as candy and explosives, plus many more. Other glycerols are made, but most of them are not classified as hazardous materials.

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Ethers

The ethers are a group of compounds with the general formula R-O-R'. The R, of course, stands for any hydrocarbon backbone, and the R' also stands for any hydrocarbon backbone, but the designation R' is used to indicate that the second hydrocarbon backbone may be different from the first. In other words, both the hydrocarbon backbones in the formula may be the same, but the "'" is used to indicate that it may also be different. R-O-R as the general formula for the ethers is also correct. The fact that there are two hydrocarbon backbones on either side of an oxygen atom means that there will be two hydrocarbon names used.

The simplest of the ethers would be ether that has the simplest hydrocarbon backbones attached; those backbones are the radicals of the simplest hydrocarbon, methane. Therefore, the simplest of the ethers is dimethyl ether, whose formula is CH_3OCH_3. Dimethyl is used because there are two methyl radicals, and "di-" is the prefix for two. This compound could also be called methyl methyl ether, or just

plain methyl ether, but it is better known as dimethyl ether. It is an easily liquified gas that is extremely flammable, has a relatively low ignition temperature of 66°F, and is used as a solvent, a refrigerant, a propellant for sprays, and a polymerization stabilizer.

The next simplest ether is the ether with the simplest alkane as one of the hydrocarbon backbones and the next alkane, which is methyl ethyl ether. Its molecular formula is $CH_3OC_2H_5$. It is a colorless gas with the characteristic ether odor. It has a flash point of 31°F, and an ignition temperature of only 374°F. This property, of course, makes it an extreme fire and explosion hazard.

The next simplest ether is actually the one most commonly referredto as "ether". It is diethyl ether, whose molecular formula is $C_2H_5OC_2H_5$, sometimes written as $(C_2H_5)_2O$. This ether is the compound that was widely used as an anesthetic in many hospitals. One of the hazards of all ethers, and particularly diethyl ether because of its widespread use, is that once ethers have been exposed to air, they possess the unique capability of adding an oxygen atom to their structure and converting to a dangerously unstable and explosive organic peroxide. The peroxide-forming hazard aside, diethyl etherhas a flash point of -56°F and ignition temperature of 356 °F; it is a colorless, volatile liquid with the characteristic ether odor. In addition to its use as an anesthetic, it is useful in the synthesis of many other chemicals, but it is an extremely hazardous material.

Another important ether is vinyl ether, a colorless liquid with the characteristic ether odor. Its molecular formula is $C_2H_3OC_2H_3$. Vinyl ether has a flash point of -22°F and an ignition temperature of 680°F. It is highly toxic by inhalation and is used in medicine and in the polymerization of certain plastics.

Ketones

The ketones are a group of compounds with the general formula R-C-R'. The -C- functional group is known as the carbonyl group or carbonyl radical; it appears in many different classes of hydrocarbon derivatives. There are only a few important ketones, and they are all extremely hazardous.

The first is the simplest, again with two methyl radicals, one on either side of the carbonyl group. Its molecular formula is CH_3COCH_3. Its proper name is propanone (propa- because of the relationship to the three-carbon alkane, propane, and -one because it is a ketone); it could logically be called dimethyl ketone, but it is universally known by its common name, acetone. Acetone is a colorless, volatile liquid with a sweet odor, has a flash point of 15° F and an ignition temperature of 1,000° F, is narcotic in high concentrations, and could be fatal by inhalation or ingestion. It is widely used in manufacturing many chemicals and is extremely popular as a solvent.

The next most common ketone is methyl ethyl ketone, commonly referred to as MEK. Its molecular formula is $CH_3COC_2H_5$. MEK has a flash point of 24°F and an ignition temperature of 960°F. It is a colorless liquid with a characteristic ketone odor. It is as widely used as acetone and is almost as hazardous.

Aldehydes

The aldehydes are a group of compounds with the general formula R-CHO. The aldehyde functional group is always written -CHO, even though this does not represent the aldehyde's structural formula. It is written in this way so that the aldehydes will not be confused with R-OH, the general formula of the alcohols. The simplest of the aldehydes is formaldehyde, whose molecular formula is HCHO. The second hydrocarbon backbone of the ketone is replaced by a hydrogen atom. Formaldehyde is a gas that is extremely soluble in water; it is often sold commercially as a 50 percent solution of the gas in water. The gas itself is flammable, has an ignition temperature of 806°F and a strong, pungent odor, and is toxic by inhalation. Inhalation at low concentrations over long periods of time has produced illness in many people. Beside its use as an embalming fluid, formaldehyde is used in the production of many plastics and in the production of numerous other chemicals. The next aldehyde is acetaldehyde, a colorless liquid with a pungent taste and a fruity odor. Its molecular formula is CH_3CHO. It has a flash point of -40°F, an ignition temperature of 340°F, and is toxic by inhalation. Acetaldehyde is used in the manufacture of many other chemicals. Other important aldehydes are propionaldehyde, butyraldehyde, and acrolein.

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Peroxides

The peroxides are a group of compounds with the general formula R-O-O-R'. All peroxides are hazardous materials, but the organic peroxides may be the most hazardous of all.

Esters

The esters are a group of compounds with the general formula R-C-O-O-R'. They are not generally classified as hazardous materials, except for the acrylates, which are monomers and highly flammable. Few of the rest of the class are flammable. There are some esters that are hazardous.

Amines

The amines are a group of compounds with the general formula R-NH_2, and all the common amines are hazardous. As a class the amines pose more than one hazard, being flammable, toxic, and, in some cases, corrosive. The amines are an analogous series of compounds and follow the naming pattern of the alkyl halides and the alcohols; that is, the simplest amine is methyl amine, with the molecular formula of CH_3NH_2. Methyl amine is a colorless gas with an ammonia-like odor and an ignition temperature of 806°F. It is a tissue irritant and toxic, and it is used as an intermediate in the manufacture of many chemicals. Ethyl amine is next in the series, followed by propyl amine, isopropyl amine, butyl amine and its isomers, and so on.

REFINERY OPERATIONS

Overview of Refinery Operations

Refining breaks crude oil down into its various components, which are then selectively reconfigured into new products. The complexity of operations varies from one refinery to the next. In general, the more sophisticated a refinery, the better its ability to upgrade crude oil into high-value products. All refineries perform three basic steps: separation, conversion and treatment. Modern separation involves piping oil through hot furnaces. The resulting liquids and vapors are discharged into distillation towers. Inside the towers, the liquids and vapors separate into components or fractions according to weight and boiling point. The lightest fractions, including gasoline and liquid petroleum gas (LPG), vaporize and rise to the top of the tower, where they condense back to liquids. Medium weight liquids, including kerosene and diesel oil distillates, stay in the middle. Heavier liquids, called gas oils, separate lower down, while the heaviest fractions with the highest boiling points settle at the bottom. These tarlike fractions, called residuum, are literally the "bottom of the barrel." The fractions now are ready for piping to the next station or plant within the refinery. Some components require relatively little additional processing to become asphalt base or jet fuel. However, most molecules that are destined to become high-value products require much more processing.

It is the ultimate objective of a refinery to transform the fractions from the distillation towers into streams (intermediate components) that eventually become finished products. This also is where a refinery makes money, because only through conversion can most low-value fractions become gasoline. The most widely used conversion method is called cracking because it uses heat and pressure to "crack"

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Chapter 5
MASS SEPARATION EQUIPMENT

INTRODUCTION

Absorption is defined as the process of taking of molecules of one substance directly into another substance. Absorption may be either a physical or a chemical process. Physical absorption depends on the solubility of the substance absorbed, and chemical absorption involves chemical reactions between the absorbed substance and the absorbing medium. Adsorption is a process involving the adhesion of molecules to the surfaces of solids, as opposed to absorption, in which the molecules actually enter the absorbing medium. Charcoal, for example, which has a great surface area because of its porous nature, can adsorb large volumes of gases, including most of the poisonous ones, and thus is used in gas masks and filters. Solvent extraction is defined as a process aimed at the separation of the components of a solution by treatment with an immiscible solvent in which one or more of the components of the solution are soluble. Membrane technology involves the spontaneous passage of a liquid from a dilute to a more concentrated solution across an ideal semipermeable membrane that allows passage of the solvent but not the dissolved solids (solutes). These four technologies or unit operations are aimed at mass separation, relying on chemical properties and concentration driving forces which drive the economics of each process. What we mean by this statement is explained by way of an example. In many situations, for example, distillation is a cost effective means for the separation of components, but it is also true that solvent extraction is sometimes practicable where ordinary distillation is uneconomical or actually impossible. Acetic acid can be recovered from a dilute solution of acetic acid in water by distillation or by extraction with a solvent like toluene, in which the acid is soluble and the water is insoluble. In contrast, close-boiling petroleum fractions, differing chemically, may be almost impossible to separate by distillation but are easily separable by means of solvent extraction.

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This chapter provides a very elementary overview of the engineering basics of four important mass separation techniques. Fundamental concepts are stressed rather than actual equipment, largely because equipment and applications are often specific as opposed to off-the-shelf or standardized designs. The techniques are based on widely different separation techniques than those that rely on physical means, such as filtration, settling, centrifugation, and others. Physical separation technologies and equipment are discussed in Chapter 6. The subject matter treated presently is

well established, and indeed modern day engineering principles, particularly absorption and extraction, were established nearly 70 years ago for applications in the petroleum and chemical processing industries. The fundamental engineering concepts developed in the 1930's through early 1950's are still quite applicable to today's designs and continued to be used.

ABSORPTION EQUIPMENT

Absorption is an old but still widely used technique that is used in the selective removal of a gaseous contaminant or product from a gas mixture. The constituent may be removed because of its economic value, or as a stage in the preparation of some compound, or because it is an air pollution problem. Removal is often effected by absorption in a liquid in which only the gas concerned is soluble. It is sometimes also necessary to free a liquid from dissolved gases. This can be achieved by contact with an insoluble gas, in which case the process is known as stripping (the reverse of absorption). Common examples of these processes are found in the absorption of sulphur trioxide in the manufacture of sulphuric acid and oleum, the scrubbing of plant exit gases to free them from hydrogen sulphide or sulphur dioxide before discharging them into the atmosphere, and the stripping of carbon dioxide from solution in water.

The objective of absorption equipment is to provide intimate contact between the liquid concerned and gas, thus facilitating the absorption process. When a large volume of gas is absorbed per unit volume of liquid, a marked rise in temperature may result and devices for the removal of the heat evolved are then an important feature of the equipment. Apart from packed towers, the types of equipment described here include agitated vessels, bubble-plate towers, cooler-absorbers and spray towers.

Agitated Vessels: These are vessels in which the gas is introduced directly into the liquid and is mechanically dispersed by means of a suitable agitator. They are used mainly where the absorption is accompanied by a slow chemical reaction between the dissolved gas and some constituent of the liquid, and sufficient time has to be allowed for this reaction to proceed to the required extent. Alternatively, they may be used where close control of the process is required in order to prevent the formation of undesirable by-products. Agitated vessels are suitable for batch operations. If used for a continuous process, counter-current flow cannot be obtained within individual vessels, and it is then almost always necessary to use a series of vessels arranged in multiple stages. The pressure-drop through agitated vessels is high compared with that for packed towers, and where this is an important factor, the use of a series of packed towers, through which a proportion of the liquid is circulated repeatedly, might be preferable.

Bubble-Plate Towers: These towers resemble the classical bubble-cap columns used for distillation. They consist of a number of shallow plates or trays over each of which the liquid flows in turn on its way down the tower. The gas enters at the bottom of the tower and is made to flow through a number of bubble caps on each plate. These caps may be of various shapes, though they usually take the form of inverted cups, and their edges are slotted so that the gas escapes from them into the liquid in the form of bubbles. The layout of a typical plate is illustrated Figure 4 of Chapter 4 (page 168). The illustration shows the arrangement of bubble caps on a plate together with the risers through which the gas enters the bubble caps and the downcomers which carry the liquid from plate to plate. The inlet weir assists in distributing the liquid over the plate, while the outlet weir maintains the desired depth of liquid. Bubble-plate towers may be preferred to packed towers when:(a) the liquid rate is so low that a packed tower could not be used effectively since the packing would not be adequately wetted; (b) when a difficult absorption duty is required, i.e. the composition of the gas at either end of the tower approaches that corresponding to equilibrium with the liquid. While faulty gas and liquid distribution can generally be tolerated in a packed tower, they may so much reduce its efficiency under these conditions as to prevent the desired performance from being achieved; (c) there is a risk that solid matter may be deposited. Bubble-plate towers, which can be fitted with manholes, are more easily cleaned than packed tower configurations; (d) a slow chemical reaction occurs in the liquid. As with agitated vessels, the use of a bubble-plate tower allows time for the chemical reaction to proceed to the desired extent.

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The pressure-drop through bubble-plate towers is high compared with that for packed towers and is of the order of 1 to 2 in. of water per plate. For difficult absorption duties where the pressure-drop available is limited, the use of a combination of packing and bubble plates may lie considered- In this arrangement a few plates are installed at the end of the tower where the composition of the gas approaches that corresponding to equilibrium with the liquid. The performance of bubble-plate towers is expressed in terms of plate efficiencies. These indicate the extent to which equilibrium between gas and liquid is attained on the plate. Though high efficiencies have been achieved with bubble-plate columns in distillation, it is important to realize that plate efficiencies in absorption or stripping may be less than 10% if the gas concerned is of low solubility.

Cooler Absorbers: When the absorption of a gas is accompanied by the evolution of heat, an important function of the absorption equipment is the removal of the heat generated. This may be accomplished by using a number of towers in series, the liquid from each tower being circulated through an external cooler. There are different types of cooler-absorbers in which processes of this type can be carried out in a single unit. The materials of which these cooler-absorbers are constructed should be of high thermal conductivity and resistant to corrosion by the substances used in the process. As an example, in the manufacture of hydrochloric acid of the

highest possible concentration, for which purpose the liquid must be cooled as absorption proceeds, the materials used include glass, tantalum and impervious carbon. Glass condenser units, consisting of a number of layers of pancake glass coil joined together in series and fitted into a length of standard glass pipe, have been used in this process. Another type of cooler-absorber also used in the production of hydrochloric acid is made of impervious carbon and is fitted with transverse tubes carrying the cooling medium, with the absorbing liquid flowing downward over the outside of the tubes and the gas flowing upward. Other types, such as stoneware tourills and silica S-bends, once widely used for the same process, are now regarded as obsolete, but may still be found in use in parts of the former Soviet Union today.

Spray Towers: A spray tower consists of an empty shell into the top of which the liquid is sprayed by means of nozzles of various kinds; the droplets thus formed are then allowed to fall to the bottom of the tower through a stream of gas flowing upwards. The use of sprays appears to offer an easy way of greatly increasing the surface area exposed to the gas, but the effectiveness of the method depends on the production of fine droplets. These are difficult to produce and suffer from the disadvantage that they are liable to entrainment by the gas even at low gas velocities. The surface area may also be reduced as a result of the coalescence of the droplets first formed. As a consequence of these effects, the large increase in surface area expected may not be achieved, or if achieved may be accompanied by serious entrainment and internal circulation of the liquid so that true counter-current flow is not obtained. A single spray tower is suitable for easy absorption duties. For difficult duties, a number of towers in series can be used.

Packed Towers (also known as wet-scrubbers): A packed tower is a vessel that is filled with some suitable packing material. The liquid is introduced at the top of the tower and flows down through the packing to the bottom, and in so doing, exposes a large surface area to contact with the gas. The gas may be introduced at either end of the tower since the gas and liquid flows may be either co-current or counter-current. The counter-current operation is most commonly used.

Numerous types of packing have historically been employed ranging from lumps of coke or quartz to complicated manufactured ceramic and plastic shapes. The arrangement of a typical packed tower is illustrated in Figure 1.

The role of the packing material is to provide a large surface area, and must have a geometry that provides large void space when packed in order to allow for good fluid flow characteristics. Raschig rings are a common example, consisting of hollow cylinders having an external diameter equal to the length of the object. Other shapes include Berl saddles, Intalox saddles, Lessing rings, cross- partition rings, spiral-type rings, and drip-point grid tiles. These objects are usually dumped into the column randomly, or if regularly shaped packing is used, stacked in an orderly fashion.

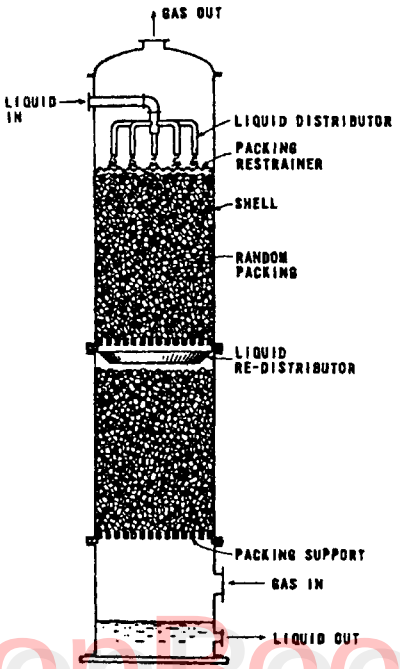

Figure 1. *Packed tower wet scrubber.*

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In general, randomly dumped packing has a higher specific surface contact area and a higher gas pressure drop across the bed will be experienced. Stacked packing arrangements have an advantage of lower pressure drop and higher possible liquid throughput, but generally are more expensive to construct. The effectiveness of a packed tower depends on the availability of a large, exposed liquid film. Hence, if poor liquid distribution prevents a portion of the packing from being irrigated, then part of the tower becomes ineffective. Liquid mal-distribution can result from improper introduction of the liquid at the tope of the tower and channeling within the tower. The liquid rate must be sufficient so as to provide sufficient wetting of the packing material, but not so great as to flood the tower. Important terms to note are load point and flood point. These terms are used to define tower capacity. For a given packing and liquid rate, a plot of gas pressure plotted against gas velocity on a logarithmic scale, identifies two break points where the slope of the curve increases. At low gas velocities the curve is nearly parallel to that obtained with dry packing, but above the break points, pressure drop increases more rapidly with increased gas velocity. The lower of these two breaks is known as the load point and the higher one is the flood point. As gas velocity increases above the load point, the liquid holdup in the packed bed increases until, at the second break point (the flood point), most of the void space in the tower is filled with liquid and liquid

entrainment in the gas stream occurs. There is excessive pressure drop during this operation. Columns should seldom be operated above the load point, however in practice towers generally operate between 40 and 70 percent of the flood point. In general, flooding velocities are considerably higher for stacked packing arrangements than for dumped packing.

Venturi Scrubbers: These devices are discussed elsewhere in the volume, but a few comments are worthy within this subject entry. Like spray towers and spray chambers, equipment that relies on the venturi principle are mainly used for removing particulate matter from gas streams, however there are some applications in gas absorption. With gas absorbers based on the venturi principle, the required interface contact is achieved by differences between the gas and liquid droplets, and by the turbulent nature of the venturi throat. Dispersion in a venturi scrubber is achieved by first injecting the liquid into the gas stream as it passes through the venturi, or by admitting the gas to the liquid stream as it passes through the venturi. In the latter scenario, the venturi also serves as a vacuum-producing device and inspirates the gas into the venturi throat. In both arrangements, a gas-liquid separation chamber is used to prevent entrainment. Venturi systems are able to achieve a high degree of liquid-gas mixing, but have the disadvantage of a relatively short contact time which generally leads to poor absorption efficiency. However, for gas systems with high solubilities and proper selection of the scrubbing liquid, the venturi can be an excellent devise. A major disadvantage is the high pressure drop with attendant high poor requirements needed for operation. In all types of equipment used for gas absorption, an essential feature is that the gas and liquid are brought into direct contact. If absorption is then to take place, the soluble constituent of the gas mixture must be transferred from the gas phase to the liquid phase. This is brought about by the diffusion of the soluble gas through the gas phase to the gas/liquid interface and hence into the liquid phase. Absorption is a diffusion process.

The driving force bringing about the diffusion arises from the difference in concentration of the soluble gas in the two phases. Just as the passage of an electric current through a circuit is opposed by the resistance of that circuit, the transfer of material by diffusion is opposed by diffusion resistance. The diffusion of the soluble gas will encounter resistance in both the gas and liquid phases. The so-called "two-film theory" provides a simple picture of the processes taking place which, while probably not corresponding closely to reality, usually provides a satisfactory basis for the design of absorption towers. The theory assumes that when a gas and a liquid are brought into contact, the main bulk of both fluids is in turbulent flow and thus uniformly mixed. Adjacent to the interface, however, and on either side of it, there are assumed to be thin films of gas and liquid in streamline flow. It is further assumed that the resistance to the diffusion of the soluble gas are located wholly in these thin films and that the main bulk of both fluids together with the interface itself offers no resistance. The principle is illustrated in Figure 2.

The rate at which the process of absorption takes place can be expressed by the following relationship:

$$\text{Rate of Absorption} = \text{Absorption Coefficient} \times \text{Surface Area} \times \text{Driving Force}$$

Expressions of this type can be written for both gas and liquid films in which the absorption coefficients are the gas- and liquid-film coefficients, respectively. The driving force across the gas film is given by the difference between the actual partial pressure of the soluble gas and that at the interface, while the driving force across the liquid film is given by the difference between the concentration of the soluble gas at the interface and that in the main bulk of liquid.

For an infinitesimal portion of the interfacial area dA at any point in the tower the two equations are:

$$dW = k_g \times dA \times (p_g - p_i)$$

$$dW = k_l \times dA \times (c_i - c_l)$$

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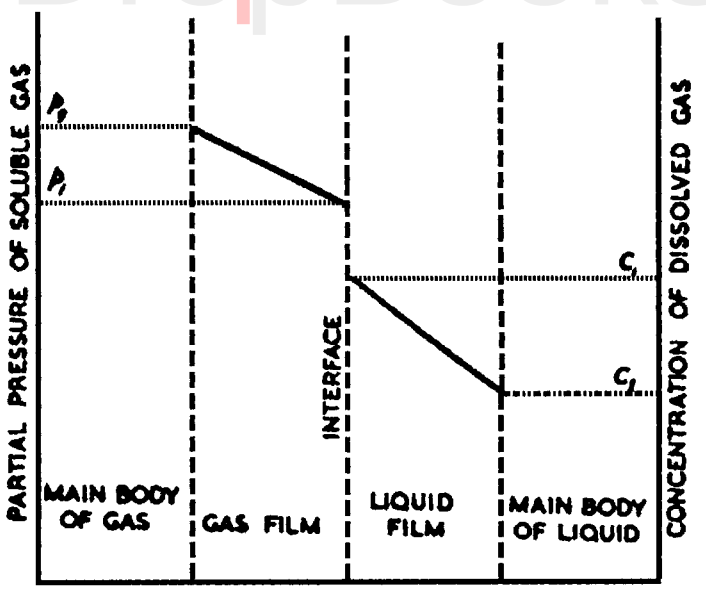

Figure 2. *Two-film theory.*

where: W = rate of absorption

k_g = gas-film coefficient

k_l = liquid-film coefficient

A = interfacial area

p_g = actual partial pressure of soluble gas

p_i = partial pressure of soluble gas at interface

c_i = concentration of soluble gas at interface

c_l = actual concentration of soluble gas in liquid.

The use of these equations is not convenient, since conditions at the interface are difficult to determine. Overall coefficients are therefore introduced by means of which the rate of absorption across both films together may be calculated. These overall coefficients can be defined in the following two ways depending upon the manner in which the overall driving force is expressed: (1) On the gas-phase basis - the overall driving force is then the difference between the actual partial pressure of the soluble gas p_g and the partial pressure p_i corresponding to equilibrium with a liquid of concentration c_l; (2) On the liquid-phase basis - the overall driving force is then the difference between the concentration c_g corresponding to equilibrium with gas of partial pressure p_g and the actual concentration of the soluble gas c_l. Thus, equations for the rate of absorption across both films together can be stated as follows:

$$dW = K_g \times dA \times (p_g - p_l)$$

$$dW = K_l \times dA \times (c_g - c_l)$$

where K_g = overall gas-phase coefficient

K_l = overall liquid-phase coefficient

p_l = partial pressure of soluble gas in equilibrium with liquid of concentration c_l

c_g = concentration of soluble gas in equilibrium with gas of partial pressure p_g

Overall coefficients cannot be predicted directly from the physical properties and flow rates of a system but must be derived from the individual film coefficients. It is important therefore to be able to relate overall and film coefficients. This can be done if it is realized that the coefficients are in effect conductances. The diffusional resistance of the two films are equal to the reciprocals of the film coefficients, and

by the usual law of the additivity of resistance in series, the sum of these reciprocals is equal to the reciprocal of the overall coefficient. Allowance has to be made, however, in adding the film resistance for the fact that partial pressures and concentrations are not measured in the same units. A factor called the solubility coefficient H is therefore introduced to allow for this difference of units. The equations relating the film and overall coefficients are:

$$1/K_g = 1/k_g + I/Hk_l$$

$$1/K_l = H/k_g + 1/k_l$$

where H = solubility coefficient, i.e. the ratio of change in concentration of dissolved gas to change in equilibrium partial pressure. According to Henry's law

$$c_l = \text{Constant} \times p_l$$

For cases where this law holds, the solubility coefficient H will be constant provided that the temperature remains constant, otherwise its value will vary with concentration. The overall coefficient can therefore change as the absorption process proceeds. Packed towers historically have been the most widely used configuration in an absorption tower. The function of tower packing is, basically, to provide intimate contact between the gas or vapor phase and the liquid phase with a minimum resistance to flow through the tower, with the purpose of providing mass transfer between the two phases. Distillation involves the transfer of mass from the liquid to the gas phase, while absorption involves mass transfer from the gas to the liquid phase. Absorption is practiced for two primary reasons, one is to recover valuable products by removal from the gas stream. For example, in petroleum production, natural gasoline is removed from well-head gas systems by absorption into a special hydrocarbon oil. Absorption is also practiced in industrial chemical manufacturing as an important operation in the production of chemical compounds. For example, in the manufacture of hydrochloric acid, one step in the process involves the absorption of hydrogen chloride gas in water. The second basic type of absorption process is the removal of a contaminant from the gas stream allowing purification of the gas or vapor. In air pollution control, this process is often called scrubbing. Among the gaseous air contaminants most commonly controlled by scrubbing are sulfur dioxide, hydrogen sulfide, hydrogen chloride, chlorine, ammonia, oxides of nitrogen and light hydrocarbons. Gas absorption, or scrubbing, is a diffusional process involving the transfer of gas molecules into the liquid phase. Concentration differences between the two phases at the points where they came into intimate contact serves as the driving force. The chemical industry

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has utilized packed towers for over 100 years. In the early stages of development, towers were filled with such available materials as broken glass, broken crockery, random threaded nuts, or other non-uniform shaped, inert materials giving a large void space when packed. Rock and gravel were used in some cases, but had the major disadvantages of being very heavy and having small surface area, giving poor performance and fluid flow capacity. Also, they were at times, not chemically inert. Lumps of coke were also used eliminating some of the weight disadvantages. The coke due to its porosity, had large surface area per unit volume. However, the exposed surface area was not completely effective as the pores are so small that they become filmed over by the scrubbing liquid, or solvent, reducing the effective surface area. These packings, all had the disadvantage that performance could not be accurately predicted, and depended on the characteristics of the packing materials obtained in each case. Therefore, no two operations performed the same in early designs.

In 1915, the development of Raschig Rings, by Frederick Raschig, gave packed tower columns a degree of predictability and dependability by providing cylindrical sections, having diameter equal to the length available, allowing varying degrees of packing openness, and flexibility with packings which provide much higher capacity and performance than the original uniformly shaped packing. Raschig Rings have been made to suit various diameters over the years with regard to performance and pressure drop. There are a wide variety of tower Raschig Rings. Tower packings have developed to the point where they now provide serious competition to mass transfer applications previously limited to trays. Usually tower designs are such that the flow is countercurrent, with the liquid introduced at the top of the tower and therefore flows down through the packing, while the gas is introduced at the bottom of the tower and flows upward through the packing material. This gives maximum "driving force" for the transfer of the solute into the solvent throughout the entire column. For this reason, this type of scrubber design is commonly used to provide high-efficiency removal of gaseous contaminants. Also, this design is most effective in removing low solubility gases. From an economic viewpoint this scrubber design generally results in a lower initial capital investment than the other scrubber designs available. However, this type of unit requires higher operating costs due to higher pressure drops and larger liquid flow rates than required in other packed bed designs.

In co-current flows, shown in Figure 3, the gas stream and scrubbing liquid flow in the same direction, generally from the top of the column downward. In this configuration there is initially a very high transfer of the solute into the solvent. This continually decreases during the flow downward through the column due to the continually decreasing concentration difference between the gas and liquid. Co-current scrubber designs may be considered for removing gases which have a high solubility in the scrubbing liquid. However, they are not suitable for removal of contaminants with low solubility because of the low "driving force", once some of

the solute has been dissolved into the solvent. This type of design is considered when there is limited space available for scrubber installation because it can be operated at much higher gas flow rates per unit area and with considerably higher liquid loadings. This is due to the fact that packing flooding is not a consideration since liquid and gas are flowing in the same direction. Co-current flow might be in the case of a very tall column built in two sections, both located on the ground. The second section might be designed for co-current flow as an economy measure to eliminate the need for construction of a large gas transfer pipe from the top of the first column to the bottom of the second. Also, for operations requiring extremely high solvent flow rates, co-current flow might be used to prevent flooding that could occur in counter-current operations. In a cross-flow scrubber design, as shown in Figure (4), the gas flows horizontally through a packed section with the solvent sprayed on the packing from the top. The gas flows perpendicular to the scrubbing liquid. This type of scrubber design is considered when removal of solid, or liquid particulate 5 microns in diameter or larger is required. This is accomplished by impingement of the particulate on the packing surface as the gas flows horizontally through the packing.

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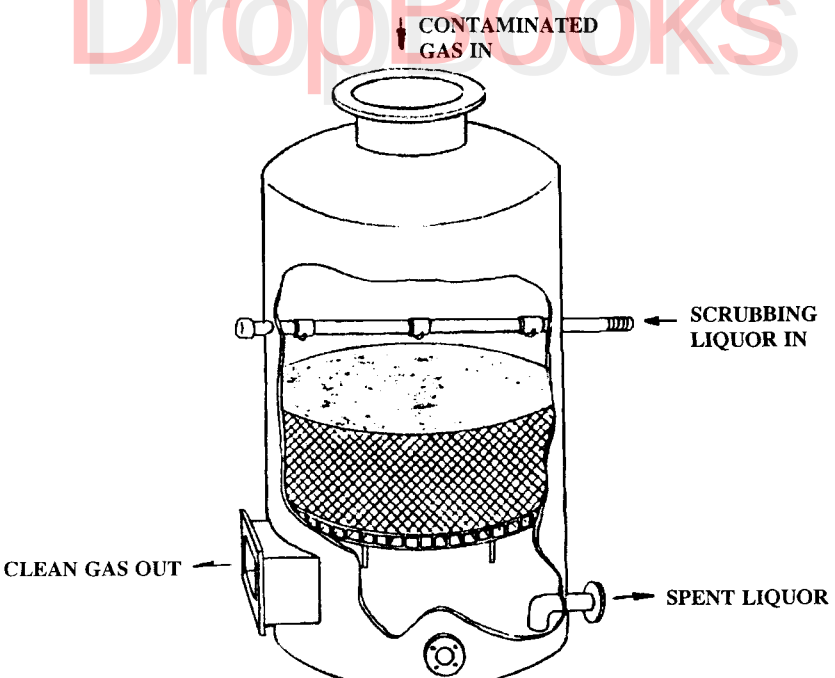

Figure 3. *Co-current scrubber configuration.*

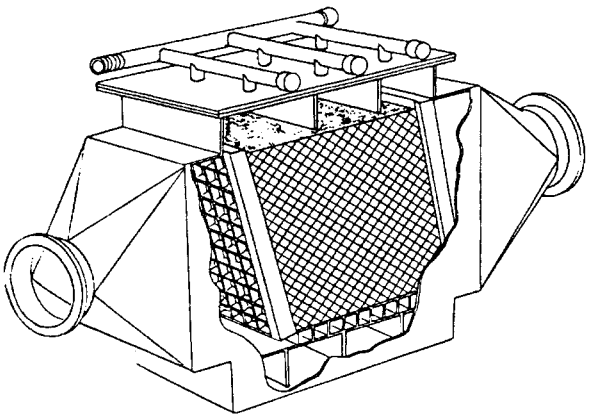

Figure 4. *Example of a cross-flow scrubber.*

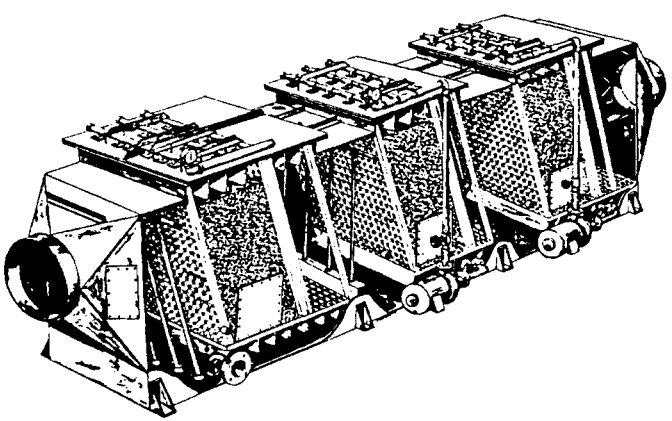

Figure 5. *Example of multiple bed cross-flow scrubber.*

These particles tend to flow in a straight line rather than following the gas stream lines around the surfaces of the packings. Once the particles hit the wet surfaces of the packing, they stick to the surface and are then flushed away by the continuing flow of scrubbing liquid. The cross-flow scrubber design is also considered when gas scrubbing, in addition to the particulate scrubbing, is required. This design offers substantial benefits when removal of several gas contaminants requires the use of several different liquid scrubbing media. By utilizing multiple, separate beds of packing, as shown in Figure 5, each bed can be wetted with a different scrubbing media, thereby preventing the liquids from mixing during the scrubbing operation.

As stated, gas absorption is a process whereby gas and liquids are brought into intimate contact and a component of the gaseous phase becomes dissolved in the liquid. With the proper selection of scrubbing liquid, the gas contaminant will have

a high solubility, allowing maximum transfer of the gas molecules into the scrubbing liquid. At the interface between the gas and the liquid streams, the velocities, in the respective phases, reduce so that a small zone, called the laminar region. It is in this zone, near the gas/liquid interface, where intimate contacting between the gas and liquid phases results. Diffusion from the laminar film of the gas into the laminar film of the liquid, is on a molecular scale, and is called molecular diffusion. A concentration difference, or gradient, between the gas and liquid, is established in the laminar interface region, and it is this concentration gradient which results in gas absorption across the interface. This diffusion does occur from the gas to the liquid and also from the liquid to the gas. A properly designed scrubber, will, of course, have a much higher transfer rate from the gas to the liquid phase. Equilibrium concentration is described as the concentrations of the solute in liquid and gas where there is equal transfer from gas to liquid and liquid to gas. Therefore, no matter how long the gas and liquid remain in contact, there will be no change in concentrations of the solute between the two phases. This rate of diffusion can be determined for gas/gas and liquid/liquid systems in the following manner: in a gas/gas steady state system, molecular diffusion of a component A from position 1 to position 2 through a stagnant non-diffusing component B is determined by:

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$$N_A = D_{AB} P (P_{A1} - P_{A2}) / RTZ_g P_{BM}$$

The diffusion coefficient can be determined from the Hirschfelder-Bird-Spotz equation as follows:

$$D_{AB} = 0.0009292T^{3/2}[(1/M_A) + (1/M_B)]^{1/2} / P\gamma_{AB}^2[\mathcal{F}(kT/\epsilon_{AB})]$$

Parameters in the above expressions are defined as follows:

NA = Rate of diffusion of component A (moles/hr/ft^2)

D_{AB} = Diffusivity (cm^2/sec or ft^2/hr)

R = Universal gas law constant (82.06 cm^3-atm/g-mole or 0.729 ft^3-atm/lb-mole)

P_{BM} = log-mean pressure of non-diffusing component B (atm)

P_A, P_B = Partial pressure of component A and B (atm); subscripts 1,2 indicates interface and equilibrium, respectively

T = Absolute temperature (°K)

Z_g = Distance in direction of diffusion (cm)

γ_{AB} = Molecular separation at collision (Å)

k = Boltzman's constant (1.38×10^{-16} ergs-°K)

ϵ_{AB} = Energy of molecular interaction of A and B (ergs)

M_A, M_B = Molecular weights of components and B, receptively

The collision function, $\mathscr{F}(kT/\epsilon_{AB})$, is given in Figure 6. Values of ϵ/k (in units of °K) and (Å) for common gases are tabulated in Table 1.

The diffusivity coefficient for a liquid-liquid system can be estimated from the Wilke correlation:

$$D_{AB} = T/\mu_B F$$

where μ_B = Viscosity of the solvent (cps)

F = Function of the molecular volume of the solute. Correlations for this parameter are given in Figure 7 as a function of the parameter ϕ, which is an empirical constant that depends on the solvent characteristics. As points of reference: for water, $\phi = 1.0$; for methanol, $\phi = 0.82$; and for benzene, $\phi = 0.70$. The two-film theory is convenient for describing gas-liquid mass transfer where the pollutant solute is considered to be continuously diffusing through the gas and liquid films.

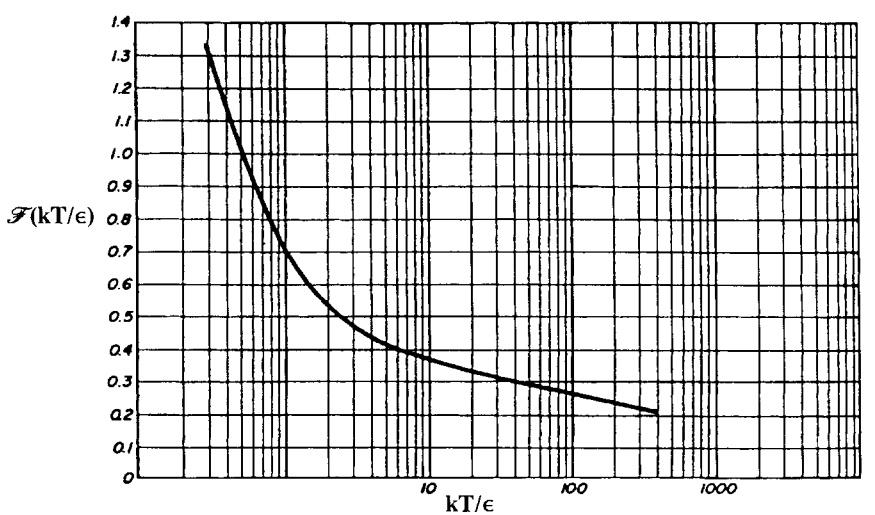

Figure 6. *Plot of the collision function.*

Table 1. *Values of Force Constants and Collision Diameters*

Gas	ϵ/k (°K)	γ_o (Å)
Air	97.0	3.617
Ammonia	315	2.624
Argon	124	3.418
Benzene	440	5.570
Carbon Dioxide	190	3.996
Carbon Monoxide	110.3	3.590
Carbon Tetrachloride	327	5.881
Diphenyl	600	6.223
Ethane	230	4.418
Ethanol	391	4.455
Ethyl Ether	350	5.424
Ethylene	205	4.232
Fluorocarbon F-12	288	5.110
Helium	6.03	2.70
n-Heptadecane	800	7.923
Hydrogen	33.3	2.968
Hydrogen Chloride	360	3.305
Iodine	550	4.982
Methane	136.5	3.882
Neon	35.7	2.80
Nitrobenzene	539	4.931
Nitrous Oxide	119	3.47
Nitrogen	91.5	3.681
Nitrogen Dioxide	220	3.879
n-Octadecane	820	7.963
n-Octane	320	7.451
Oxygen	113.2	3.433
Propane	254	5.061

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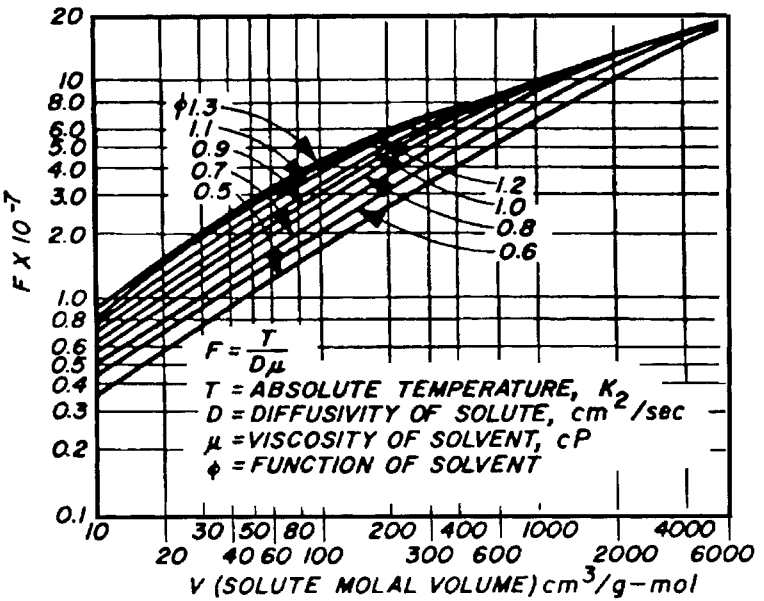

Figure 7. *Solvent characteristic, F, versus solute molal concentration.*

The driving force for the absorption process is the concentration difference between the gas and liquid phases, which in terms of the interface, can be expressed by partial pressures:

$$N_A = k_g (P_A - P_{Ai}) = k_\ell (c_{Ai} - c_A)$$

k_ℓ and k_g are the liquid and gas phase mass transfer coefficients, which can be determined from the following relationships:

$$k_\ell = D_{ABg} P / RTZ_g P_{BM}$$

$$k_g = D_{AB\ell} / Z_\ell$$

Since it is difficult to measure interfacial concentrations of the gas and liquid film (P_{Ai} and c_{Ai}) and the distance in direction of diffusion (Z_g and Z_ℓ), this problem can be eliminated by using the over-all mass transfer coefficients, K_ℓ and K_g, for the liquid and gas respectively. The rate of absorption of the solute depends on the concentration difference or gradient of the solute between the liquid and gas phases. This concentration gradient can also be expressed in terms of the difference between

the bulk concentrations, and the equilibrium concentration described earlier. This equilibrium concentration may be defined by:

$$P_A^* = Hc_A$$

where H is Henry's law constant. When the equilibrium concentration is substituted for the interfacial concentration, then the local mass transfer coefficients can be expressed in terms of the overall mass transfer coefficients as follows:

$$N_A = k_\ell(c_{Ai} - c_A) = K_\ell(c_A^* - c_A)$$
$$= k_g(P_A - P_{Ai}) = K_g(P_A - P_A^*)$$

The relationships between the local and overall mass transfer coefficients are:

$$1/K_g = 1/k_g + H'/k_\ell$$
$$1/K_\ell = 1/k_\ell + 1/(H'k_g)$$

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Note that H' is simply Henry's constant corrected for units. When the solute gas is readily soluble in the liquid solvent, Henry's law constant (H or H') is small and K_g approximately equals k_g, and the absorption process is controlled by the gas film resistance. For systems where the solute is relatively insoluble in the liquid, H is large and K_ℓ approximately equals k_ℓ and the absorption rate is controlled by the liquid phase resistance. In most systems, the solute has a high solubility in the solvent selected, resulting in the system being gas film resistance controlled.

For any particular system, a graph can be constructed using the concentration of the solute in the liquid phase (c_A) and the concentration or partial pressure of the solute in the gas phase (P_A) as the abscissa and ordinate, respectively. A line indicating the equilibrium concentrations of the solute in the gas and solvent drawn on this graph, results in an equilibrium diagram.

Generally, these concentrations are expressed in terms of moles of solute per mole of pure solvent (liquid phase) and moles of solute per mole of inert gas (gas phase), thus making the material balance calculations easier.

A typical equilibrium diagram is shown on Figure 8. As noted earlier, a counter-current flow scrubber provides the highest efficiency of operation. The respective concentrations and rates, as shown in Figure 9 provide the basis for a material balance across this scrubber design.

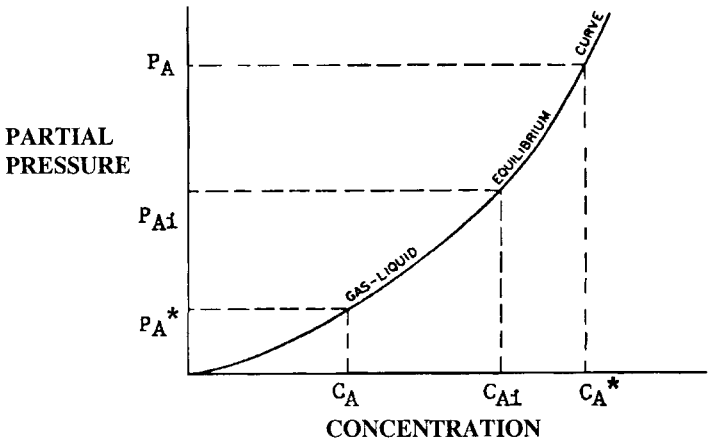

Figure 8. *Gas-liquid equilibrium curve.*

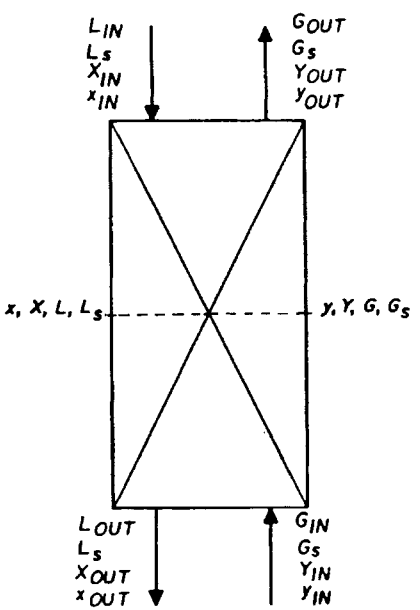

Figure 9. *Material balance diagram.*

Defining the following variables for the purposes of a material balance:

$$X = x/(1 - x); \; Y = y/(1 - y)$$

From Figure 9, a material balance across the scrubber section shown is as follows; providing a basis for a graphical method of tower sizing:

$$G_s = (Y_{in} - Y_{out}) = L_s(X_{out} - X_{in})$$
$$Y_{in} - Y_{out} = (L_s/G_s)\{X_{out} - X_{in}\}$$

L_s and G_s are the pure liquid and inert gas loading rates, respectively, in units of lb-moles/hr-ft^2. The second expression is the operating line on an equilibrium diagram. In all scrubbing application, where the transfer of solute is from the gas to the liquid, the operating line will lie above the equilibrium curve. When the mass transfer is from the liquid to the gas phase, the operating line will lie below the equilibrium curve. The latter case is known as "stripping".

The development of the operating line is required to determine the minimum liquid flow rate requirement for scrubbing. In an actual scrubber design, all of the following information will be known: the concentration of the solute in the gas (Y_{in}), the concentration of the solute in the entering solvent (X_{in}), the entering gas flow rate (allowing calculation of G_S), and the required concentration of the solute in the exit gas (Y_{out}). To calculate the minimum liquid flow rate (L_s), concentrations Y_{in} and X_{out} will be at equilibrium (on the equilibrium line at the point Y_{in}, X_{eq}). The slope of the operating line between X_{eq}, Y_{in} and X_{in}, Y_{out} represents the condition resulting in minimum required flow rate, as shown on Figure 10.

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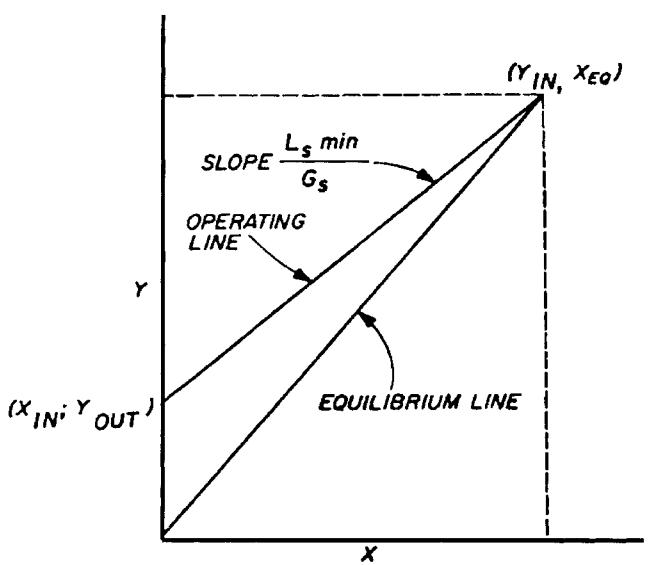

Figure 10. *Operating line for absorption.*

The slope of the line is L_S/G_S. This procedure provides the minimum theoretical solvent requirements. In practice, however, liquid rates are selected at least 25 to 100% greater than the theoretically calculated minimum rate. Typically, packing manufacturers suggest that liquid rates between 2 and 5 gpm/ft^2 result in the best operation. Higher liquid flow rates will effect the degree to which the packing surfaces become fully wetted. Obviously, higher liquid flow rates will result in a greater portion of the packing surface being wetted, thereby giving higher active liquid gas contacting area and a higher rate of mass transfer. When approached from an economic standpoint, higher liquid flow rates results in a higher efficiency of scrubbing per unit tower volume, but lower concentration of the solute in the solvent. This results in a higher quantity of scrubbing liquid used, which generally means a higher operating cost, especially where solvents are valuable. However, this design approach results in minimum initial capital investment. Where lower liquid flow rates are considered, the result is larger scrubbing towers, resulting in higher initial capital investment cost, but lower operating usage of scrubbing liquid. Scrubbing manufacturers advise that above a certain liquid flow rate, known as "minimum wetting rate" (MWR), there is virtually no increase in mass transfer rate. This is due to the fact that at this rate, the full effective surface area is wetted by the scrubbing liquid. In the selection of a solvent for any particular scrubbing system, several aspects of the scrubbing liquid should be considered. These factors are: (1) the solubility of the gas should be relatively high to enhance the rate of absorption and to decrease the quantity of solvent required; (2) the solvent should have relatively low volatilities to reduce solvent loss due to evaporation; (3) the solvent should be as non-corrosive as possible to reduce the construction and maintenance costs of the equipment; (4) the solvent should be inexpensive and readily available; (5) the solvent should have relatively low viscosity to increase tower capacities and reduce flooding potentials; (6) the solvent should be non-toxic, non-flammable, chemically stable and have a low freezing point.

The diameter of a packed tower, or the gas mass flow rate per square foot of tower area, is governed either by the limiting pressure drop through the scrubber, based on blower pressure drop capacities, or on limitations on liquid build-up in the packing and/or entrainment caused by the gas. On a log-log plot of pressure drop versus mass flow rate in pounds per hour per square foot as shown in Figure 11, a plot of the data for dry-packing shows a straight line with a slope of two. As liquid flow rate is increased, the pressure drop increases. This is due to the reduction in available flow area, due to accumulation of liquid onto the packing. As the gas flow rate is increased, there is a break in the slope of the curve. This break point is called the loading point, labeled A'_1. Operation above the loading point is not normally recommended due to the increased pressure drop encountered. Further increases in the gas velocity result in a second break in the slope of the curve. This break point is called the flood point, labeled A'_2. At this condition, liquid becomes the continuous phase within the packing with the gas bubbling through.

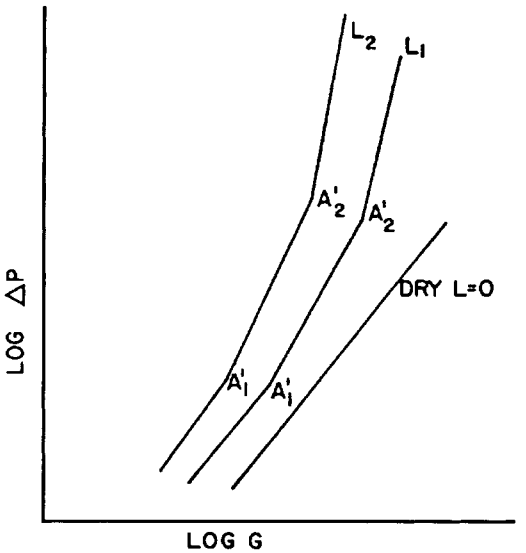

Figure 11. *Loading and flooding conditions.*

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In addition to the large increases in pressure drop, high liquid entrainment into the exiting gas stream also occurs.

The packing factor (F) is an important design variable in determining the capacity of any packing design from a flooding, or pressure drop standpoint. This-factor was first described by Sherwood in 1938. The empirical correlation is:

$$F = a_p/\epsilon^3$$

where a_p is the specific area of the packing (ft^2/ft^3) and ϵ is the packing material void fraction. It has been found, however, that this approach does not take into account the effects of variations of the active area of the packing due to variations in liquid loading and packing characteristics, such as surface shielding from adjacent packing pieces, wettability, and wall and support effects. Most packing factors supplied by the various manufacturers are calculated to fit the observed flooding curves for the packing design under consideration. Further work by Eckert (see references) showed that the packing factor varied substantially for varying ratios of liquid/gas flow rates, and also for the operating pressure drop at each flow rate. Where data, such as that described by Eckert is available, they should be used for design purposes. However, published packing factors are generally adequate for most design purposes, except that there is some tendency to overdesign the tower

at high liquid/gas flow ratios, especially when Raschig Rings are used. The packing factor (F) is used in conjunction with Figure 12 to determine the estimated gas flow rates at flooding and at different pressure drops. There is no correlation establishing an estimate of the loading condition, however, for most commercially available packings, it is safe to design for gas flow rates 40 to 70% of the flooding rates calculated. Calculation of required tower diameter, based on the above flooding flow rate design criteria, can be accomplished utilizing Figure 12.

To perform the calculation, first calculate the abscissa value. Then, utilizing the flooding line curve, the ordinate value can be obtained. By solving for G, the flooding gas flow rate can be determined. By choosing an appropriate percentage of flooding gas flow rate, the tower diameter can be calculated. Alternatively, where the operating velocity is fixed by the allowable pressure drop, the appropriate pressure drop curve in Figure 12 can be used to determine the ordinate value which can then be solved for G. The ratio of operating velocity to flooding velocity can then be determined using the same procedure. The required height of the packed section of a scrubber is governed by the removal performance of the solute required of the system. In this plot, ψ is the ratio of the density of water to the density of the scrubbing liquor, μ is the solvent viscosity, ρ_L and ρ_G are the densities of the liquid and gas, respectively.

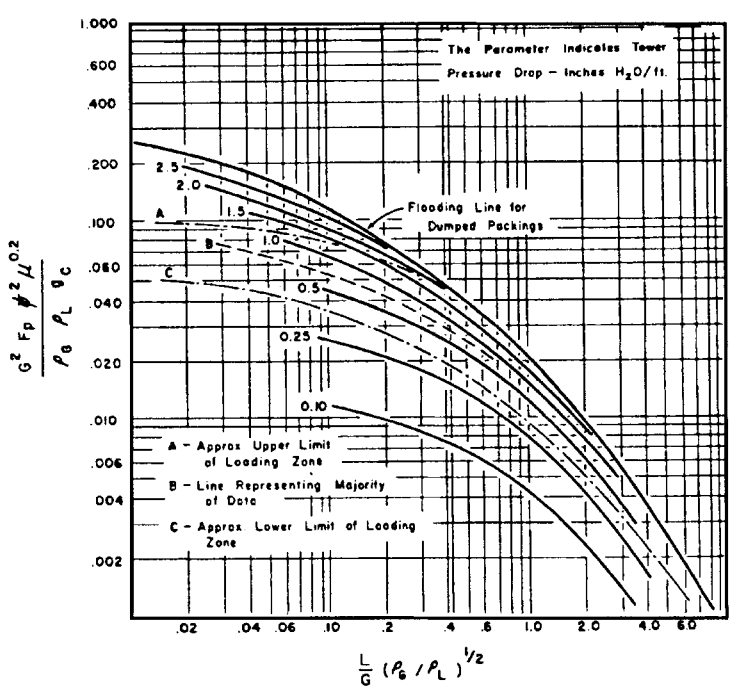

Figure 12. *Pressure drop - velocity plot.*

Going one step further, the required height of packing is governed by the rate at which absorption of the solute into the solvent takes place. This will be a function of the packing size, shape, liquid rate and liquid distribution over the surface of the packing. The height (Z) of packing required is determined by the following relation:

$$Z = H_{OG} \times N_{OG}$$

where H_{OG} is the height of a transfer unit (ft), and N_{OG} is the number of overall transfer units required. Performing a material balance over a small section of packing for the counterflow scrubber case (system definition given in Figure 13), we obtain the following over area dz:

$$d(G_m y) = K_g a(y - y^*)dz$$

From whence it follows that:

$$G_m dy/(1 - y) = \{[K_g a(1 - y)_m][y - y^*]dz / (1 - y)_m$$

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subscript "m" refers to a mean value.

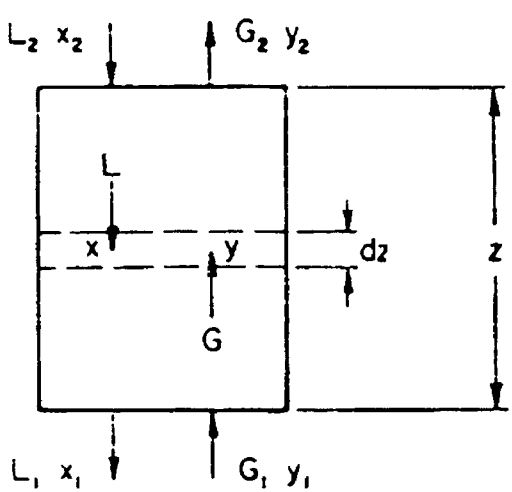

Figure 13. *Differential mass balance definition.*

Assuming that the absorption process is gas-film controlled, and that the concentration of the solute is small (i.e., $1 - y \approx 1$), then:

$$K_g = k_g$$
$$N_{OG} = N_G = \int dy/(y - y^*)$$

Introducing Henry's law ($y^* = H'x$); then the above expression reduces to the following:

$$N_{OG} = (y_1 - y_2)/(y - y^*)_{lm}$$

where the log-mean definition for concentration is as follows:

$$(y - y^*)_{lm} = [(y_1 - y^*) - (y_2 - y^*)]/[\ell n(y_1 - y^*)_1/(y_2 - y^*)_2]$$

For an absorption system in which the solute is highly soluble in the solvent, such that the equilibrium pressure of the solute over the absorbing liquid is very low, we may write:

$$y - y^* \approx y$$
$$N_{OG} = \ell n(y_1/y_2)$$

The above expression indicates that the number of overall mass transfer units, N_{OG}, is only controlled by the concentrations of the solute in the inlet and outlet gas streams.

The following tabulation of calculations shows the relationship between the number of transfer units and the percent of solute removed from the gas stream:

	% Absorbed		**% Absorbed**
N_{OG}	$100 \times (y_1 - y_2)/y_1$	N_{OG}	$100 \times (y_1 - y_2)/y_1$
1	63	4	98.2
2	86.5	5	99.3
3	95.0	6	99.97

The height of a theoretical unit (H_{OG}) can be obtained from the actual process conditions whenever possible. Tower packing manufacturers have information on a wide variety of specific packing systems that enable the determination of H_{OG}. Alternatively, this information can be estimated from values of $K_g a$ using he following relationship:

$$H_{OG} = Gm/[K_g aP(1 - y)_{ln}]$$

Most tower packing data is based on information from standard performance test systems. These systems are usually based upon an air-feed stream with 1% CO_2 solute in the inlet stream. The scrubbing liquid entering a column is 4% NaOH in water with a carbonate conversion of 25%. Tests are normally run at a liquid temperature of approximately 75 °F. When making a selection of the specific tower packing design to be used in a scrubbing system, several packing qualities should be considered before final selection. Many of these factors interrelate with one another and various trade-offs between the subjects covered below, must be considered in the overall selection of the tower packing:

1. Cost: initial capital investment is an important consideration in the selection of any tower packing design. Costs are generally expressed in dollars per cubic foot. Generally speaking, plastic packings are less expensive than metal packing materials. However, the operating costs of the packing must also be considered. These include, power consumption, packing replacement costs, when required, solvent feed rates, and operating flexibility.
2. Corrosion resistance: it is critical to select construction materials such that neither the gas and solute nor the scrubbing liquid will corrode the packing materials. Ceramic and plastic materials, are commonly selected for this reason.
3. Temperature resistance: at high temperatures, some tower packing materials, such as plastics, may deteriorate rapidly. Further, the potential loss of cooling of upstream gases, must be considered if materials susceptible to damage by heat are considered for use. An upset might require a complete repacking of the scrubber.
4. Pressure drop: while initial installation costs are important, the continuing increase in the cost of energy makes the pressure drop consideration more and more important. The continuing operating costs involved in the selection of a high pressure drop packing, may in some cases, outweigh its initial cost advantages. Rating of existing blowers, fans, etc., when existing equipment is to be used, must be within the pressure drop limits of the packing design selected.

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5. Large active exposed surface area: the specific surface area, e.g., the exposed surface area of the packing per cubic foot of packed volume, is important since larger specific area means a larger area available for contact between the liquid and the gas, resulting in more efficient scrubber operation.

6. Percent void space: high amounts of open flow area tend to reduce pressure drop, minimize the possibilities of plugging due to solids accumulation, and increase loading and flooding liquid and gas flow rates.

7. Complete utilization of the packing surface: the packing shape should be designed to prevent nesting and blockage of the surface area of one packing piece by another, thereby, reducing the effectiveness of the nested pieces for mass transfer. The packing design must also minimize liquid channeling, by forcing redistribution of the scrubbing liquid as it flows through the packing. The packing should also promote turbulent contact between the gas and liquid to allow maximum renewing of the gas and liquid at their interface. By eliminating channeling of the gas and liquid, stagnant areas, which are prone to fouling, will be eliminated.

8. Structural strength: the packing must be strong enough to withstand normal loads during shipping, installation, service, and maintenance. Also, thermal fluctuations, when present, must effect the integrity of the packing. The packing must also be strong enough to withstand the weight loads imposed by the weight of the packing resting upon it.

9. Weight: as mentioned above, weight of the packing can limit bed height. Low-weight packing, can increase the allowable packed bed depth between intermediate packing bed supports. Low side thrust loads, on the tower shell, may also be important to allow minimum tower wall thicknesses and the use of low-strength/low cost materials of construction.

10. Removal for cleaning: the tower packing should be designed for ease of removal when solids build-up may be a consideration.

11. Design flexibility: the packing should have a wide operating range with little variation in efficiency for changes in gas and liquid flow rates.

Tower packings are manufactured in a wide variety of shapes, sizes, and materials of construction. The shapes are generally broken down into two broad categories; namely, random packing and systematic packing. Random packings are installed in a non-uniform random arrangement, most often by dumping them into the retaining vessel, as described below. This type of design allows higher specific surface area, but normally results in higher pressure drop. Systematic packings are systematically stacked or laid out to provide intentionally uniform openings and surface area distribution throughout the tower packing area. These designs, in some cases, provide less surface area, but result in more uniform liquid distribution. This type

of packing design is generally more costly to install due to the substantial extra labor involved, or the higher cost of the packing itself.

Random type packings can generally be broken down into three general shape geometries: ring-type packings, saddle-type packings, and miscellaneous. The first ring type packing was the Raschig Ring, which can still be found in use in some applications, although it is relatively inefficient. Lessing rings are similar to Raschig Rings, but have a single internal cross-rib which provides strength and additional surface area for contacting. Lessing Rings are manufactured in porcelain, chemical stoneware, stainless steels, and other metals. However, with the development of newer packing designs, this packing is used on a very limited basis. Cross-partition rings are similar in geometry to the Lessing Ring, except there is a double internal cross-rib. This type of packing can serve two purposes. First, it can act as conventional tower packing. Secondly, when stacked over a series of cross bar support grids, as shown in Figure 14, in brick-lined vessels, it can act as a support for dumped packing resting upon it. These units are normally made in sizes from 3inch through 6 inch diameter from stoneware and porcelain. Larger sizes are almost always stacked. Cyclohelix spiral rings are similar to Raschig Rings, except inside the cylinder is a spiraling internal. This is normally supplied with either a single, double, or triple spiral. The purpose of the spiral is to impart a swirling, spiraling turbulence to the gas to obtain good mixing with the scrubbing liquid. This type of ring is almost always stacked rather than dumped to obtain maximum effectiveness of the spiral. Other packing designs feature openings in the walls of the ring with the resultant metal from the openings curved inward to partially simulate the cross-partition ring.

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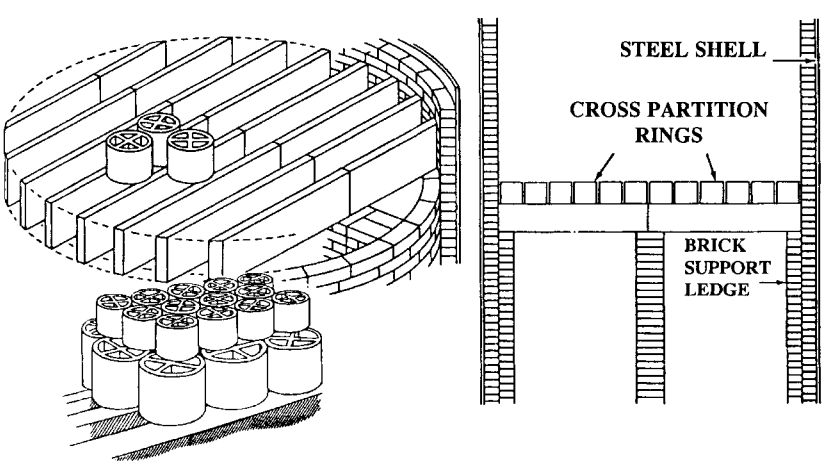

Figure 14. *Cross-partition rings used as packing supports.*

These designs, such as the Pall ring and the Flexi-ring allow much increased gas flow rates at the same pressure drop and liquid rate as the earlier Raschig Rings. This allows use of smaller diameter towers with resultant economic savings. Improved ring designs also provide higher efficiency due to a larger amount of available surface area. Also, greater gas turbulence, because of the tortuous path the gas must follow, results in better contacting.

The best known saddle-type packing is the Berl Saddle, which has the appearance of a saddle. The saddle configuration was introduced many years ago to reduce the amount of nesting and blockage of available surface area which occurred when packing vessels with Raschig Rings. Intalox saddles and Flexi-saddles are modified saddle type packing designs, improving on the basic Berl saddle configuration. Improvements in the shape further reduce the nesting or stacking which occur during installation, thereby allowing maximum surface area accessible to gas and liquid flows. These packing types are normally made in plastic, or in chemical porcelain or stoneware. Additional design variations include scalloped edges on the packing. The scallops provide many more interstitial transfer points per unit of volume than comparative packings. These transfer points continuously renew the liquid surface, greatly increasing the rate of mass transfer. The scalloped edges also serve to immobilize the packing within the bed resisting settling effects so that the initial free space and pressure drop are maintained. Helix packing are modified saddle-like packing designs normally made from polypropylene and other plastic materials. These packing designs were developed specifically for use in scrubbing type applications. Some advantages of these packing designs include: non-nesting, resulting in less channeling and increased turbulence; lower pressure drop; self cleaning design; lighter weight allows bed support savings; corrosion and heat resistant; and high mechanical strength allowing bed heights greater than 20 feet. Some miscellaneous random packing designs worth noting are the Rosette type packings, Tellerette Packing, and Spiral. Manufactured from a wide variety of plastic materials, these packing designs claim to provide many more interstitial hold-up points than can be obtained with other packing designs. These hold-up points and the use of non-wetting plastic materials create a dispersion/agglomeration cycle of the scrubbing liquor, which results in constantly fresh liquid being exposed to the gas flow.

Systematic type packings can generally be broken down into four areas, based on their internal geometries: stacked ring type, mesh type, grid type, and corrugated type. The larger diameter ring type packings can be installed by laying the rings out systematically on end, and stacking several layers one on top of the other. Ring designs used where this installation approach is considered include Raschig Rings, Lessing Rings, cross-partition rings, and cycloheilix spiral rings. This installation approach is normally used only for rings 3 inches in diameter and larger. This approach is not normally used for smaller rings because of the exponential increase in installation time required. This systematic stacking of rings requires substantial

additional labor time, resulting in much higher installation costs. Mesh and fiber-type packings have been shown to be highly efficient in absorption and distillation applications. It is in distillation, however, where these designs are most frequently used due to their extremely high performance levels. In distillation and stripping applications, the potential savings in operating costs, due to the high efficiencies, will often outweigh the packing cost. Packings are manufactured from multiple strands of wire, knitted together to form a flattened sock of mesh.

Wood grid packing designs consist of beams of wood stacked with alternate layers perpendicular to each other. This type of packing provides lower packing efficiency than the packing designs discussed to this point, however, this design is extremely economical and may be considered where low performance is required and where the wood will not be affected by the chemical environment. Drip point grids are ceramic bricks with vertical holes similar in shape to cinder blocks. When stacked in a scrubber, they provide a large surface area and high void fraction, resulting in low pressure drop. Again, performance is lower than packing designs discussed earlier. However, drip point grids are used occasionally in applications requiring particulate removal. Glitsch grids are a high open area, low pressure drop packing design, manufactured from stacked layers of perforated metal sheets which are cut to fit the vessel inside diameter. This design is considered in systems which have high solids loading as they are not susceptible to solids plugging. This packing design is frequently used in distillation columns where coking is a potential problem. It also finds use in some particulate scrubbing applications.

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Corrugated packings are manufactured from relatively inexpensive metal or plastic corrugated sheets. Sheets are laid parallel to the flow with the corrugations running at a predetermined angle to the flow. Corrugations between adjacent layers of the sheet are reversed in direction, providing maximum mixing of the gas and liquid flow. The overall packing is manufactured in short length elements along the direction of the f low. These elements are stacked so that the direction of the corrugated sheets are 90° to the previous pack, creating complete mixing of the fluids over several elements. The depth of the corrugations can be varied. Higher capacity and lower pressure drop are obtained with deeper corrugations while shallower corrugations provide higher contacting efficiency. This type of packing is precut to fit inside the vessel diameter required.

Once the total bed depth of packing has been determined, the tower must be laid out so that the maximum bed depth is no more than 8 tower diameters or 20 feet maximum. Liquid distributors, to ensure uniform liquid distribution and well designed support plates, are required to realize the full potential of the packing in any application. Depending upon the size and nature of the packing being considered, the material is either stacked uniformly or dumped randomly. Where packings are installed by dumping, it is preferred that the tower be packed wet, to assure maximum randomness of the packing elements, and minimum breakage where ceramic packings are used. At least four feet of water should be kept above

the packing surface at all times for the above reasons. If possible, the water level should be up to the loading manway. When packing a large tower, the packing should be dumped at random spots, rather than developing a slope or cone packing surface, as a cone surface would tend to force the liquid flow outward to the vessel wall, creating poor liquid distribution. To get the best performance from the tower packing, it should not be shaken, dropped, walked on, or handled excessively after installation since any breakage will reduce the void area, and the efficiency. Top hold-down of the packed bed is generally recommended for plastic, ceramic, or carbon tower packing to prevent the upper portion of bed from becoming fluidized. If the packing is not held down, the resulting breakage tends to further restrict flow and encourages further surges which will ultimately destroy the packing. In this case, the hold-down rests directly on the bed and prevents bed fluidization based solely on its weight. The hold-down is not attached to the wall as this would provide no restraint once the bed had settled. For metallic packing, a bed limiter is normally substituted for the hold-down. This acts only to retain the packing inside the packing area, and to prevent it from being carried out of the vessel due to its light weight, at high pressure drops. Bed limiters are always attached to the walls of the scrubber, since breakage and crushing are not problems with these construction materials. With no moving parts, tower packing requires little maintenance during normal operations. Corrosion or solid particulate accumulation are generally the problems encountered, thereby requiring periodic replacement.

Packed tower wet scrubbers to a certain degree are considered an outdated technology, although there are still many industrial operations that utilize them throughout the world. The greatest application for scrubbers from an environmental standpoint is likely flue gas scrubbing. Many systems are packaged designs offered by vendors. An example of a packaged system is illustrated in Figure 15. In this case the scrubber cleans 97-100% of the sulphuric dioxide (SO_2, causing acid rain) and hydrochloric acid gases out of the flue gas, and also a high percentage of the dust. Since all the sulphur in the flue gas is removed, there are more options when selecting oil quality, as it becomes possible to bum waste oil and oils with a high sulphur content. The flue gas is cooled down towards 30 °C making it possible to reduce the oil consumption by up to 20% when using the scrubber. The amount of savings depends on the plant's work efficiency throughout the year and on how effectively the low temperature heat is used. The scrubber is simple, but efficient: a liquid flow meets the hot flue gas and particles, dust and water-soluble gases are washed out. The water is neutralized, cooled and reused. The water in the gas condenses when the hot gas meets the cold liquid. This process aids the removal of particles from the flue gas. In order to reduce the acidic content of the gas, lye is added to the water used in the scrubber. The salts formed can be released into the sea where they are found naturally. The scrubber is profitable for larger boiler houses for central heating, district heating, and for boilers of water or steam in industry. It is the oil consumption and the operating time, which determine the

profitability of the scrubber. The oil consumption ought to be larger than 500 tons/year in order to have a profitable scrubber plant. Standard scrubber packages that are commercially available are for oil boilers with an effect from 1.5 MW to 70 MW.

The trend in the use of gas scrubbers, particularly on small units, has shifted towards the use of non-packed columns. These are basically venturi scrubbers that were briefly described earlier in this chapter. Based on the operating principle of a liquid jet ejector, gas scrubbers are an economical means of resolving problems caused by the harmful nature of exhaust gases by scrubbing, purifying, neutralizing and eliminating them almost entirely from smoke, together with polluting odors. In some cases, the gases are a too high temperature and the scrubbing fluid can be used for cooling them. For the purification of gases having a strong smell, the scrubbing fluid has to include a chemical solution capable of interacting with the treated gas. A simplified diagram of a venturi gas scrubber is illustrated in Figure 16, in which there are three important design features, namely, the scrubbing fluid inlet configuration, the gas inlet, and the spray jet. The materials of construction used are chemically resistant, and are usually made from stainless steels, plastics, graphite, or other suitable materials.

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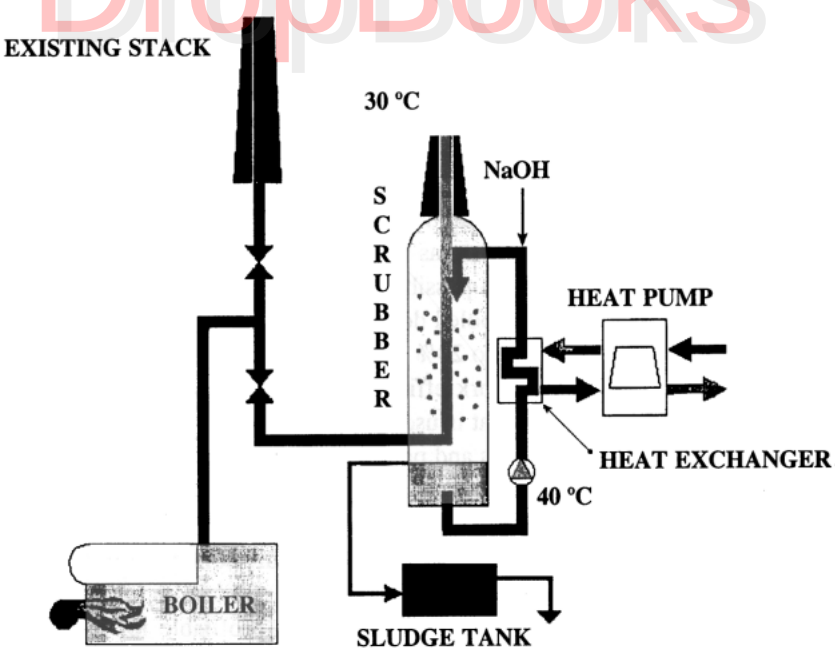

Figure 15. *Example of a packaged flue gas scrubber system.*

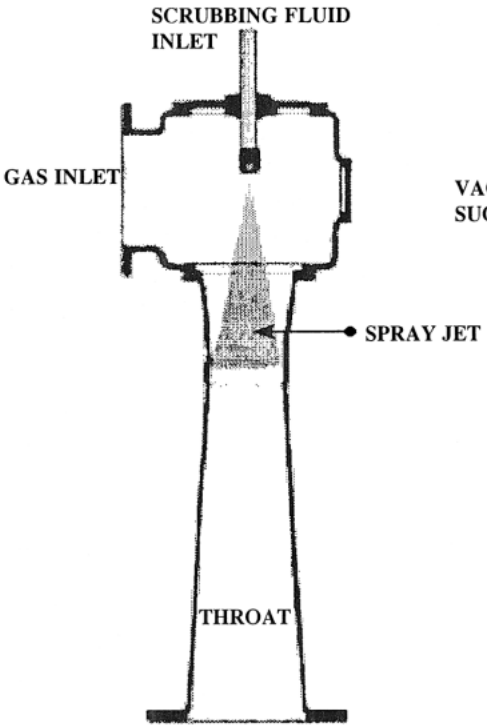

Figure 16. *Venturi scrubber.*

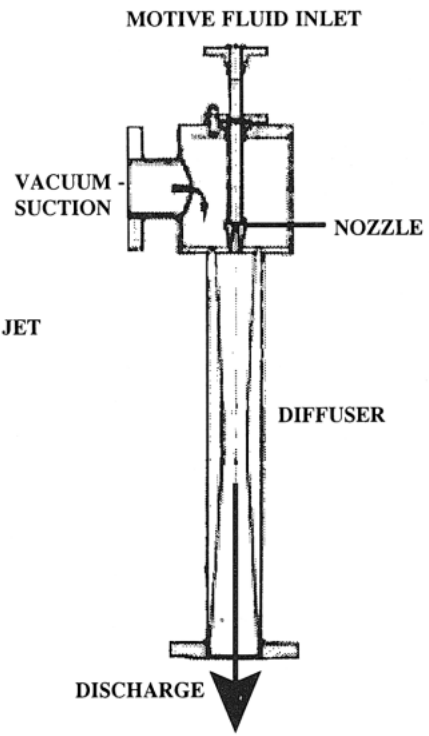

Figure 17. *Jet vacuum pump.*

Note than an ejector is the generic name of a jet appliance capable of aspirating different products: gases, liquids and solids (powders, granulates or sludge) and takes different names according to its functions: jet vacuum pump, thermocompressor, gas scrubber, eductor, etc. The operating theory is the same for every type of ejector. In the case of a jet vacuum pump, the apparatus is capable of compressing gases and fluids (optimum compression ratio: 7 to 8 = ratio between discharge pressure and suction pressure) so as to obtain constant suction pressure (more or less intense vacuum, depending on process requirements). An example of a jet vacuum pump is shown in Figure 17. Gas absorption is a unit operation that chemical engineers will certainly encounter numerous times throughout the chemical process industries. The principles outlined in this chapter provide a basic understanding of the classical design principles and practical operating guidelines. It is not uncommon for systems to be field erected for specialty applications, although again, the trend is to purchase modular or packaged units for applications such as fume scrubbing or flue gas scrubbing application. Section A of the chapter references can be consulted for additional information.

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Chapter 6
MECHANICAL SEPARATION EQUIPMENT

INTRODUCTION

There are several types of equipment used for either processing intermediate products, product separation or recovery, for improving feed and product stream qualities and purities, and for pollution control purposes - that separate phases making use of the physical properties differences between chemical components. These equipment categories may be used to separate solids or particulate matter from liquids, slurries and gases, to concentrate solids by removing moisture through means other than by the direct application of thermal energy, to segregate solid particulate matter by size or density differences, to separate liquids by means other than solvent extraction or distillation. These equipment are broadly classified as mechanical devices or physical separation equipment. They capitalize on properties such as density, viscosity, surface tension, electrostatic forces, size, and several other properties, whereby the differences between the components, and in some cases the phases of components are significant. These property differences then are the driving forces which form the intended applications and design basis for such equipment. The equipment is so generalized in application that they represent basic unit operations and standard pieces of equipment that are relied on in almost any industry sector. Therefore, specific examples are illustrative only, simply because these types of machinery are versatile in almost any industry sector. This chapter covers the following equipment categories: filtration, settling, floatation and clarification, centrifugation, particle segregation and classification. Emphasis in this chapter is given towards providing discussions on fundamental operating principles and equipment specification criteria. It is an overview chapter, meaning that theory and heavy design practices are not addressed, but rather, sufficient information and additional references, including vendor contacts and Web sites are provided for establishing preliminary equipment specifications. The vast majority of these equipment have been in use in various allied industry applications for more than a century, and hence design criteria are well established based on many tried and proven prototypes and industry applications. Many of the operations and equipment described are often applied in either preliminary, intermediate or

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final stages of a process operation, and less often as the primary process step. Classical references sited at the end of this chapter can be consulted for theoretical considerations.

FILTRATION EQUIPMENT

Gas Cleaning Equipment

Fabric filters, more commonly called baghouses or dust collectors, have been in use since the early 1900's in the mining industry. Today's applications extend throughout the chemical process industries with considerable emphasis in industrial air pollution control. Dry dust filters are available in sizes ranging from a few square feet up to several hundred thousand square feet of cloth. Gas flows that can be handled by individual units range from under 100 cfm to over 1,000,000 cfm. The fabric filter's design is similar to that of a large vacuum cleaner. It consists of bags of various shapes constructed from a porous fabric. Filter bags are available in two major configurations, namely, flat (envelope) bags and round (tubular) bags. Figure 1 illustrates the operation of a baghouse. The dust-laden gas enters the module through an inlet diffuser that breaks up the gas stream and evenly disperses the dust. The heavier dust particles settle into the hopper and the fine particles rise through the tube sheet into the bags.

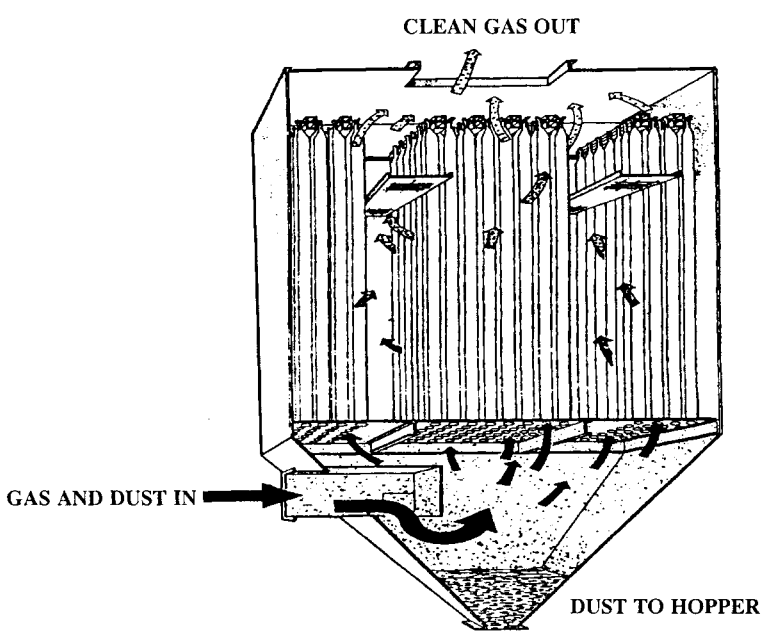

Figure 1. *Basic operation of a baghouse.*

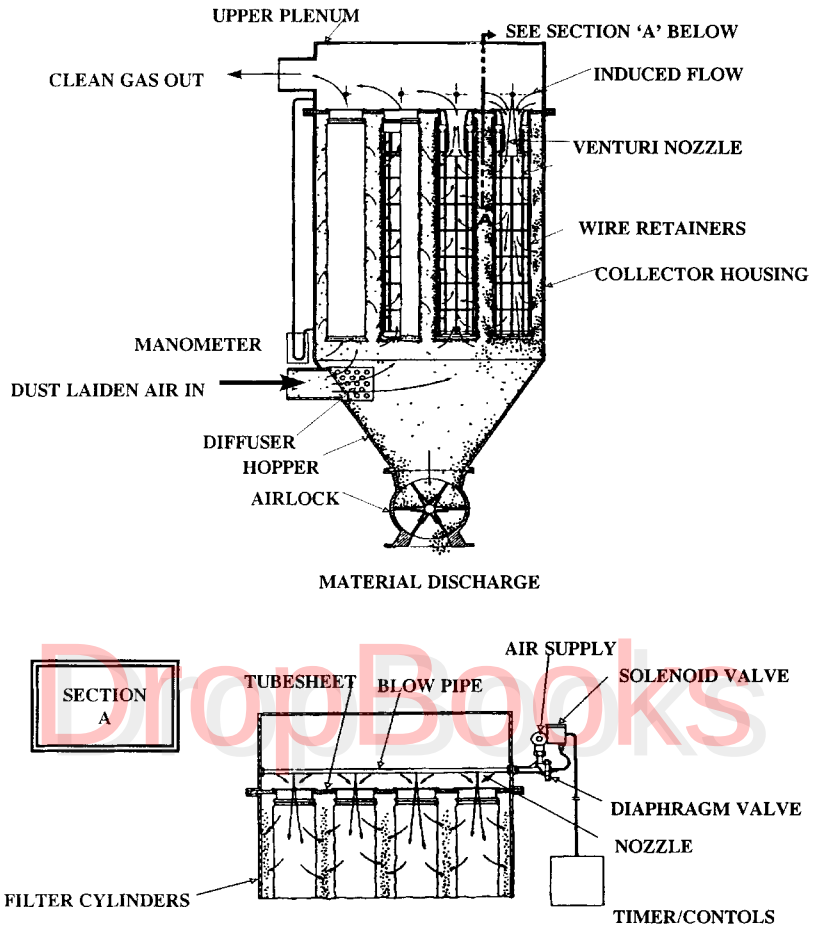

Figure 2. *Design features of a baghouse.*

Particles typically as small as 0.5 microns in diameter are collected on the inside of the bags, while the cleaned gas passes through the fabric. Dust is removed from the bags by periodic shaking accomplished by an automatic shaker. The frequency of cleaning depends on the type of dust, the concentration, and the pressure drop which must be overcome. The dust shaken from the bags falls into the hopper below and is removed by a rotary airlock, screw conveyor, or other devices. Figure 2 illustrates some of the specific design features and components of a baghouse operation. Series modules can be joined to provide any desired capacity. When two or more modules are joined together, a single module can be shut down for bag cleaning and then returned to service. The simple closing of an inlet or outlet damper diverts the dirty gas stream to other modules. Thus, the gas is filtered continuously. Multi-module installations typically employ a large single fan or small

individual fans mounted on each module. Small fan arrangements are more flexible and eliminate the need for outlet ductwork and the foundation that are required for a large fan. Individual fans often simplify maintenance and permit fan, motor, drive, or other components to be changed readily without interrupting normal service. Any one module can be shut down and isolated from the rest of the system while still maintaining full operation and efficiency levels. The particles to be removed play an important role in the selection of a fabric and filter efficiency. Specifically, particle density, concentration, velocity, and size are important. Each of these properties is interrelated to the pressure drop of the system, which has a direct impact on operating or particle capturing efficiency. Principal variables directly related to pressure drop are gas velocity, the cake resistance coefficient, the weight of cake per unit area, and the air-to-cloth ratio. The cake resistance coefficient is dependent on the particle size and shape, range of the particle sizes, and humidity. Weight of the cake per unit area is related to the concentration of particulate matter.

Large scale air filtration systems for air pollution control applications are generally packaged systems. Commercially available packages are available which include both the reverse air and compressed air types. Reverse air baghouses have typical diameters that range from 8 to 18 ft, from 6,000 cfm to 94,000 cfm. Compressed air baghouses usually include both bottom bag removal (i.e., through bin vents) as well as top bag removal units. Commercial systems can be round or square/rectangular in configuration. Round (plan view) body sections are usually retrofitted with square plenum sections.

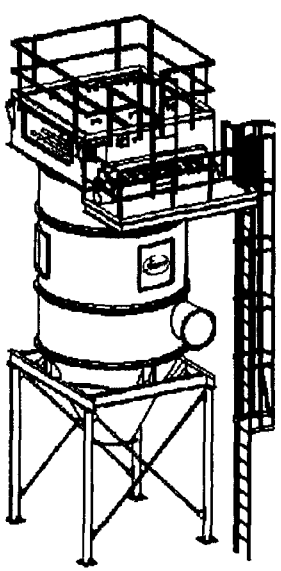

Figure 3. *Round design.*

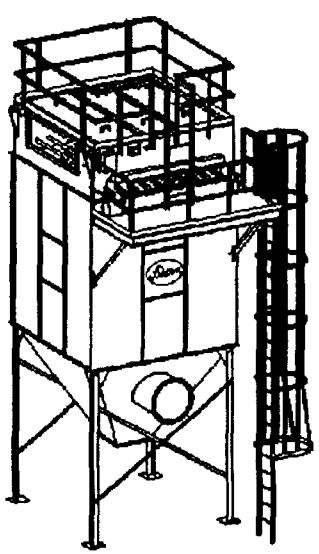

Figure 4. *Square design.*

Depending on the accessibility afforded by adjacent equipment, aged ladders and access platforms are often quoted as options by vendors. Square body configuration systems (plan view) have square body and plenum sections. Safety handrails surrounding the plenum are standard. Structural supports are usually optional, the height of which depends on system location. Figures 3 and 4 provide sketches of the two basic configurations.

The design of a fabric filter system must satisfy two criteria, namely, high partcile capture efficiency and low pressure drop. Prediction of the pressure drop and knowledge of its dependence on operating conditions of the filter are necessary for proper design. Fabric filters composed of thicker felt materials have a complex orientation of fibers which can achieve a high collection efficiency with less dust buildup. Consequently, felt filters cannot be cleaned by mechanical shaking due to embedment of fine particles. Instead, a high pressure air stream (60 to 120 psi) is used to disengage the dust particles. The specific application dictates the type of fiber. For example, fiberglass filters can withstand higher temperatures than wool; nylon is a poor fiber to use for chemical resistance. The fabric is designed to withstand thermal, chemical, and mechanical action.

Fabric filters can be categorized according to the particular cleaning method, the filter capacity, the type of filter media, the temperature capability, and the type of service (either intermittent or continuous). There are three major cleaning methods employed: shakers, reverse air, and pulse jet. The oldest and most widely used cleaning method is mechanical shaking. The casing is divided into an upper and lower portion by a tube sheet. The woven fabric tubular bags are located in the upper portion with a pyramid-shape hopper in the lower end. Each bag is supported between a flexible cap and a fixed thimble. Gas velocity entering the hopper is reduced, causing the coarse particles to settle out. The gas enters the tube on the inside causing the fine particles to be collected, with the clean gas passing through the fabric into a common outlet manifold. After a certain amount of dust buildup, the flexible support mechanically shakes the particle loose from the fabric into the hopper. The reverse air cleaning baghouse operates in the same manner as the mechanical shaking arrangement, except a reverse air flow replaces the shaking process. An air vent located in the outlet manifold is opened allowing atmospheric air to enter the casing, thereby collapsing the bags and dislodging the dust particles. Baghouses incorporating the pulse jet cleaning method are constructed with an upper and lower compartment separated by a tube sheet. The upper portion serves as the discharge manifold. The felted filter bags are supported by a venturi-shaped thimble attached to the tube sheet. A compressed air jet is located above each filter bag to facilitate cleaning. Internal frames (mesh cages) with a closed bottom prevent the collapse of the bags during the cleaning cycle. Dirty gas enters the hopper and is then directed into the casing, passing through the filter bags. The dust is collected on the outside surface, allowing the clean air to pass through the fabric and out the discharge manifold. The filter bags are cleaned by the force of the pulse jet expanding the bags (see Figure 5).

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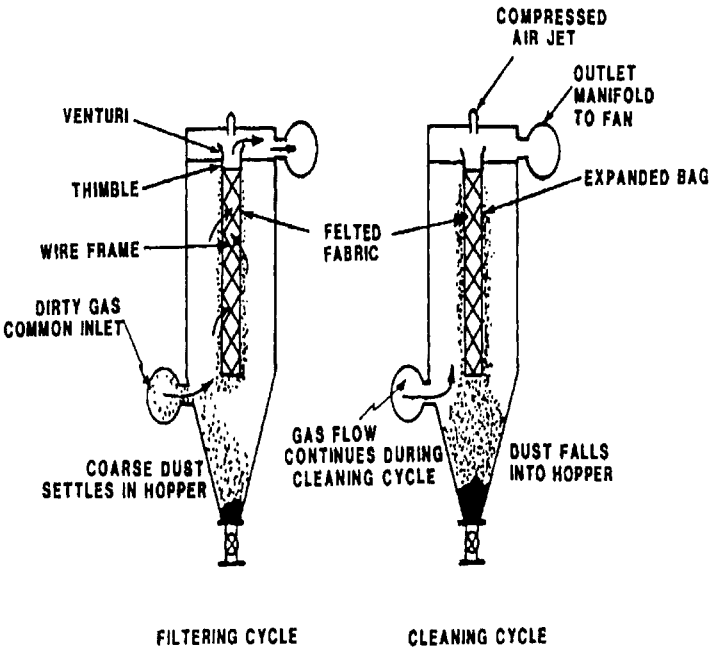

Figure 5. *Automatic baghouse with pulse jet cleaning.*

Filters are grouped according to the capacity by volume as follows: small volumes (i.e., below 10,000 acfm), medium volumes (i.e., 10,000 to 100,000 acfm), and large volumes (i.e., >100,000 acfm). The filter-media types include woven and felted media. Temperature capabilities of the media range from higher temperatures (> 400 °F), to medium temperatures (200 to 400 °F) and low temperatures (< 200 °F). The ability of the fabric to collect fine particles and maintain a good cleaning process should serve as the basis in selection of a fabric. As the dust layer or so-called filter cake layer builds up, flow resistance increases. Cleaning reduces the gas flow resistance and maintains the proper pressure drop across the filter.

Cloth filters are designed to remove three types of particles, and each type incorporates the basic principles of air filtration. Particles in the submicron size are collected as a result of the Brownian motion and bridging. As the particles build-up on the surface of the media, the collection surface areas increase causing particles to be captured. Collection efficiencies in excess of 99.95 % are possible. Particles having diameters in the 1 to 10 μ (micron) range and coarse particles (above 10 μ) rely on inertial collection. Efficiencies of 90 % to 95 % are achieved with particles under 10 μ in size. Efficiencies of coarse particles fluctuate from 50 % to 99.9 %. The reason for such variations in efficiency is a result of re-entrainment caused by gas flows at high velocities. High velocities can force the particles through the pores of the fabric.

Fabric filters are made of natural fibers, such as cotton and wool, or from synthetic fibers, depending upon their application. Cotton and wool are available in spun form, which limits the individual fibers to a few inches in length. Spun fibers can produce characteristics not found in filament (continuous) fibers. Filters composed of spun fibers are thicker, bulkier, heavier and provide a higher permeability to air flow. Synthetic fibers offer greater versatility such as higher operating ranges and corrosion resistance, but are more costly. Synthetic fibers are available as multi-filament fabrics, which are the most widely used and produced characteristics not common on spun forms. Multi-filament fabrics are light weight, of high tensile strength and high dimensional stability, abrasion resistant, and easy to clean.

Collection efficiency is affected by thread count, yarn size, and twist of the yam. Permeability is increased by decreasing the thread count (either warp or fill), and subsequently increasing the pore area. Collection efficiency is reduced by increased permeability. A balanced weave is optimum for efficient operation. Permeability is also a function of the yarn size. The larger the yarn size, the lower the permeability. Yarn diameter, which can be altered by the twist of the yarn, also affects permeability; the smaller the diameter, the higher the permeability.

The ability of the gas stream to permeate the filter is also affected by the shrinking and elongation of the fabric. Elongation of the fibers results in increased pore space, and conversely shrinkage decreases the pore volume. Fabrics of good dimensional qualities are essential to fabric life and efficiency.

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Fabric finishing steps enhance the collection efficiency of a system. The most commonly applied finishing steps include calendering, napping, singeing, glazing, and coating. Calendering is where surface fibers are pushed down onto the fabric medium by high pressure pressing on the fabric. Napping refers to scraping the filter medium, which raises the surface fibers. Singeing involves separate surface fibers that are removed by passing the filter medium over an open flame. Glazing involves surface fibers fused to the filter medium by high pressure pressing at elevated temperatures. Finally, coating involves a surface preparation over the fibers that reduces self-abrasion.

In order to specify a fabric filter, the properties of the incoming gas and particulate matter must be well defined. The properties of the gas essential to the problem definition are volume, temperature, moisture content, and acid gas concentration. The dust properties of importance are the density, particle size, particle size distribution, and the dust loading. The size of the fabric filter dust collector is directly related to the gas-to-cloth ratio as previously noted, which can be calculated by dividing the total gas volume by the total area of cloth or filter medium. Corrections to the gas-to-cloth ratio are required for the volume of air introduced to the unit and to account for the area of cloth not exposed to the incoming gas. The type of fabric selected is a function of the cleaning process employed. For example, the pulse-jet cleaning method would be used in conjunction with the felt fabric to achieve proper cleaning and efficiency. The filter medium type is the next major consideration following the gas/cloth ratio and cleaning method.

Cotton bags are used in standard installations and are the most economical. A maximum operating temperature of 180 °F is recommended for continuous use with 225 °F allowed for surge conditions. Wool bags are used for applications with dust particles of a combustible nature, or with operating temperatures of 200 °F and an allowable surge temperature of 250 °F. Nylon has a greater tensile strength than cotton or wool and provides excellent abrasion resistance. Fiberglass is most resistant to high temperatures, with a maximum operating temperature of 500 °F. To increase the allowable temperature, fiberglass filters are silicone treated to permit their use in applications such as in carbon black production plants. Replacement of bag filters generates the highest maintenance and cost of the system. Typical causes of bag failure include too high of a gas to cloth ratio, metal-to-cloth abrasion problems, chemical attack by the gas stream or particulates, inlet velocity abrasion, and excessive gas temperatures. The quality of the fabric and method of cleaning are additional factors to consider in evaluating service and maintenance costs. If a filter bag tears, it is important to repair the bag as quickly as possible to prevent abrasion to adjacent bags by jet streams of dust discharging out of the damaged bag. This type of bag failure is limited to inside bag collection types of dust collectors. The speed of repair is determined by the opacity of the outlet bag. In a compartmentalized system, broken bags can be found by monitoring the emissions while isolating one compartment at a time. To prevent a higher filter velocity, damaged filter bags within a compartment should not be replaced with clean bags. The higher velocity could create greater pressure drop or failure due to dust abrasion. An alternative is to plug or tie off the flow. To recap, the first step in selecting a fabric filter is to define the magnitude of the particulate loading. Knowledge of the particulate matter collected, properties of the gas stream, and the cleaning method are essential to proper design. Improper design leads to low efficiency and unscheduled maintenance. Prior to selection, results from a related application should be investigated. An alternative is to operate a pilot unit to ensure the most optimum gas-to-cloth ratio for a specified pressure drop. With proper design, operation, and maintenance, better than 99.9% efficiency can be achieved, depending on the application. A major advantage of fabric filters, is their ability to operate at a high efficiency at all loads from maximum down to very low gas flow. Some disadvantages of the system are the space requirements and high maintenance costs. Other problems associated with fabric filters are plugging of the fabric due to operation below the dew point or break down of the filter bags, resulting from high temperatures.

Fabric filters can be more costly to operate and maintain than electrostatic precipitators, cyclones, and scrubbers; however, fabric filters are more practicable for filtration of specific dusts. For example: fabric systems are the typical control method for toxic dusts from insecticide manufacturing processes, salt fumes from heat treating, metallic fumes from metallurgical processes, and other applications. Any other control method may not be as efficient, nor economically feasible for such applications.

Up to this point, we have described the operations and features of an industrial baghouse dust collector, which is essentially a system containing filter fabric of some kind which removes dust and particulate from a gas streams. The cleaned gas can then be vented to the atmosphere. If the dust is a by-product, the dust can be recycled back into the process. In contrast, a cartridge dust collector uses gravity together with a downward airflow pattern to provide extremely high filtration efficiency while reducing energy consumption. The cartridge is mounted in a horizontal and slightly downward sloping position. This design increases filter life and improves airflow because both gravity and airflow are pushing the dust downward into the collection hopper. Filters in baghouses typically need to be replaced after approximately one year of operation. Users should consider the cost of maintenance and filter replacement when purchasing or specifying a dust collector. This is an important cost factor to consider when comparing to cartridge filters which generally are less expensive. In addition, cartridge filters tend to be more efficient than a baghouse. A cartridge collector collects 99.9% of submicron particulate as compared to more typically 99.0% for a baghouse. This characteristic can be particularly important in applications involving toxic dust or when air is being recirculated.

On smaller filtration devices such as those used in fume control in indoor air applications, the real objective is to purify air through materials that trap contaminants which may include both particulate matter and gases. In such systems the fabric is impregnated with activated carbon or zeolite. Gases and odors filter through activated carbon and/or zeolite, an organic mineral. HEPA (high efficiency particulate air filter) is an effective medium for screening particles developed by the US Atomic Energy Commission. True medical rated HEPA is 99.97% effective in removing particles above 0.3 microns, or one three-hundredth the size of a human hair.

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Clearly the largest application for filtration is air pollution control. On a worldwide basis the annual sales revenues for all types of pollution control equipment is around 150 billion dollars, with air pollution control devices being a substantial part of these equipment sales. The following is a summary of Web sites recommended to be consulted for specific vendor information on air filtration devices. A brief description of each site is provided. Many of these sites will link the reader to other sites containing additional information on product information.

Recommended Web Sites on Product Information for fabric filters and related devices:

- **Astec Microflow**: Manufacturers of fume cupboards, fume hoods and fume cabinets. Leaders in filtration and containment technology. *http://www.astec-microflow.co.uk*
- **TMS Air Filtration Systems Limited**: Specialist in air filtration, extraction and ventilation systems. *http://www.westmids.co.uk/tmsairfiltration*

and ventilation systems. http://www.westmids.co.uk/tmsairfiltration

- **Munktell Filter AB**: Highly qualified producer of filter media in medical, industrial, environmental controlling and analytical, and purifying filtration of air, fumes and fluids. http://www.munktell.se
- **Coppys**: Provider of portable ventilators which help fume and vapor removal, compressed air filtration, personnel cooling & confined space ventilation with propane heaters, cooling mist & ventilation blowers. http://www.coppus.com
- **Airflow Systems, Inc**: Resource for the collection and filtration of dust, smoke, mist, fumes and other airborne contaminants generated during industrial and commercial manufacturing and processing applications. http.,Ilwww.air,flowsystems.com
- **Clean Air Ltd**: Fume cabinets (or cupboards, UK) designed and produced quickly and efficiently for a wide range of Research, Pharmaceuticals, Chemicals, or educational settings. http://www.cleanairltd.co.uk
- **Campbell Environmental Systems:** Sells Austin air HEPA air filters and Miele HEPA vacuum cleaners http://www.airwaterbestprices-comlindex.html
- **Air Cleaning Systems, Inc.**: Sells, services and provides equipment and parts for air filtration systems. http://www.aircleaningsystems.com
- **Conquest Equipment Corporation**: Industrial filtration products and services to clients in the wood, paper, plastics, metal, aerospace, and mining industries. http://www.conquestequipment.com
- **Hoffman & Hoffman, Inc.**: Manufacturers sales representatives in the selection and application of commercial heating, ventilation, air conditioning, filtration and DDC systems. http://www.hoffman-hoffman.com
- **Clean Air Machine Corp.**: Air purifiers - commercial site offers customized filters for specialty applications aimed at removal of microscopic particulates and gases. http://aco.ca/tibbits/index .html
- **Aircon Corp.**: Site markets air pollution systems including dust control systems and equipment. http://aircon-corporation.com
- CECO Environmental: Experts in improving air quality through the use of fiber bed filter systems, high temperature baghouse filter fabrics, scrubbing technology, and on-site air quality monitoring. http://www.cecofilters.com
- **Fortress Designs**: Supplier of dust control equipment, baghouses, dust collectors, fabric filters, dust filters, bin vents. http://www.fabricfilters.com
- **Schrader Environmental Systems, Inc.**: Air pollution control technologies such as catalytic oxidizers, packed tower wet scrubbers, dust collectors. http://www.angelfire.com

orientation, Table 1 provides an equipment selection guide that can be used in selecting the most appropriate piece of equipment for a gas cleaning application. The table is based upon general vendor literature reported in terms of removal efficiency and general applicability to a gas cleaning problem.

Table 1. *Equipment Selection Guide.*

Particle Size	**Small**	**Tiny**	**Invisible**	**Submicron**
Particle Type	>1/64", 20 mesh	4/1000", 140 mesh	4/100,000" - 8/10,000"	<4/100,000"
Size =	850 microns	51 - 850 microns	2 - 50 microns	<1 micron
Solids Particulates	Gravity settler	Gravity settler, Cylcone	Coarse (Grit) filter, Cyclone	Cartridge filter
Liquid Particulates	Cyclone	Gravity settler	Cyclone, Baghouse	Scrubber
Combination	Fume filters, Scrubber	Fume filters, Scrubber	Mist and fume filters	Scrubber

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Liquid Filtration Equipment

Liquid filtration equipment is commercially available in a wide range. Proper selection must be based on detailed information of the slurry to be handled, cake properties, anticipated capacities and process operating conditions. One may then select the preferred operational mode (batch, sem-batch or continuous), and choose a particular system on the above considerations and economic constraints. Continuous filters are comprised of essentially a large number of elemental surfaces, on which different operations are performed. These operations performed in series are solids separation and cake formation, cake washing, cake dewatering and drying, cake removal, and filter media washing. The specific equipment used can be classified into two groups: (1) stationary components (which are the supporting devices such as the suspension vessel); and (2) scraping mechanisms and movable devices (which can be the filter medium, depending on the design). Either continuous or batch filters can be employed in cake filtration. In filter-medium filtration, however, where particulate matter are retained within the framework of the filter medium, batch systems are the most common. Batch filters may be operated in any filtration regime, whereas continuous filters are most often operated under constant pressure. It is important to note that equipment used in liquid filtration applications fall into three general operational modes based on the

difference in orientation between gravity force and filtrate motion. These orientations are: forces acting in the opposite direction (i.e, countercurrent systems), forces acting in the same direction (cocurrent), and forces acting normal to each other (cross-flow).

Solid-liquid separation is a fundamental unit operation that exists in almost every flowscheme related to the chemical process industries, ore beneficiation, pharmaceutics, food or drinking water and wastewater treatment. The separation techniques are very diverse. As a unit operation, there are many types of equipment that are used, not just filtration systems. In addition to filtration by such equipment as vacuum and pressure filters described below, there are centrifugation by filtering and sedimenting centrifuges, sedimentation by conventional, storage and high-rate thickeners, clarification by conventional, solids-contact and sludge-blanket clarifiers, polishing by precoat filters, pressure and deep-bed filters, and upward separation by dissolved-air flotation .

The first group of liquid filtration equipment described is the vacuum filter. Vacuum filters are relatively simple and reliable machines, and as such they have gained wide acceptance in the chemical processing and allied industries. The various types of vacuum filters are summarized in the chart provided in Figure 6. For a sense of physical dimensions, the following are some typical sizes: drum and disc filters are up to 100 m^2 in filtration area; horizontal belt filters up to 120 m^2; tilting pan and table filters up to 200 m^2. A typical filter station is comprised of the following components and the flowsheet can look like the example provided in Figure 7:

- The vacuum filter
- One or more vacuum receivers (if separation of the mother filtrate from the wash filtrate is needed)
- One or more filter pumps (either receiver mounted or connected to the seal tank of a barometric leg)
- Vacuum pump (either liquid ring or rotating vanes type)
- Piping and instrumentation

There are several advantages that vacuum filtration has over other solid-liquid separation methods. Some of these advantages include:

- Continuous operation (with the exception of the Nutsche type filter),
- Intensive soluble recovery or removal of contaminants from the cake as accomplished by countercurrent washing operations. This is especially the case with horizontal belt, tilting pan and table filters, which are described later in this subsection,
- Polishing of solutions (as performed on a precoat filter operation),

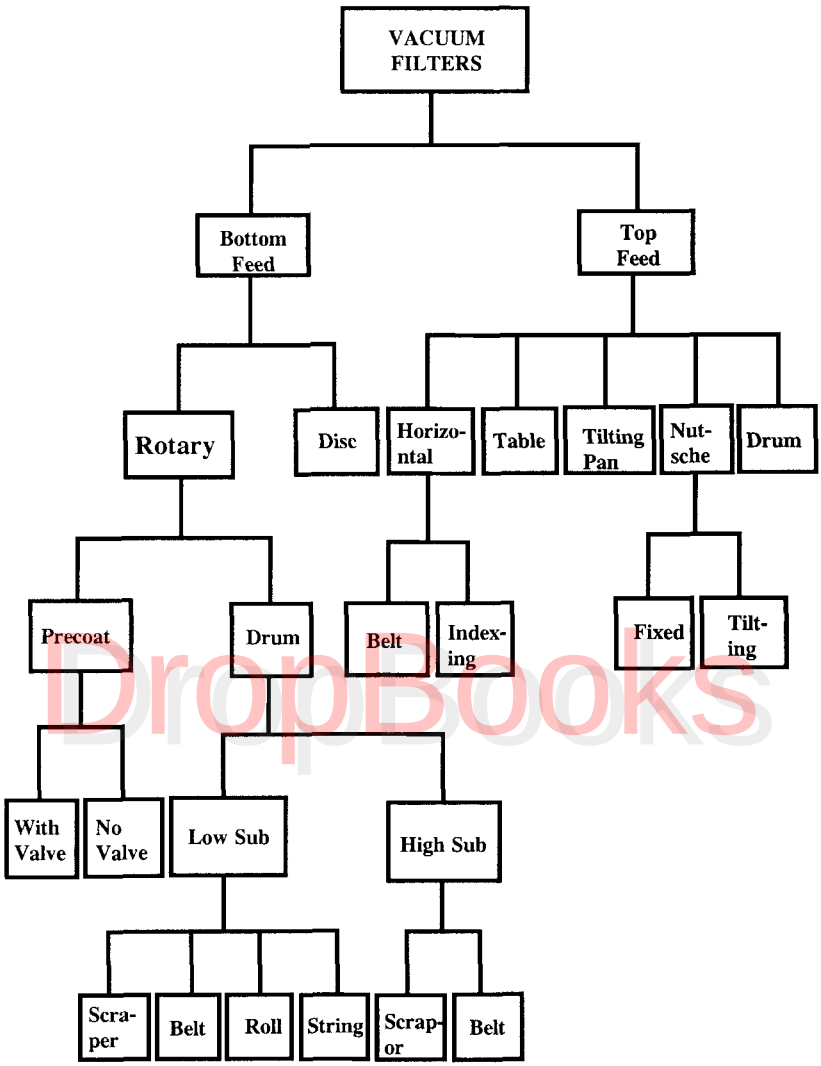

Figure 6. *Organizational chart for vacuum filters.*

Additional advantages that vacuum filters have over other separation methods is the ability ti produce relatively clean filtrates by using a cloudy port or a sedimentation basin, as in the cases with the horizontal belt, tilting pan and table filters; they provide easy access to cake for sampling purposes; they allow easy control of operating variables such as cake thickness and wash rations; commercial systems are available in a wide range of materials of construction. The principle disadvantages of vacuum filters include higher residual moisture in the cake, untight

construction so it is difficult to contain process vapors, and high power requirements as demanded by the vacuum pump.

One of the oldest filters applied throughout the chemical processing industry is the rotary vacuum drum filter, which is illustrated in Figure 8. This machine belongs to the group of bottom feed configurations. Rotary drum filters are typically operated in the countercurrent mode of operation. The principle advantage of these machines is the continuity of their operation.

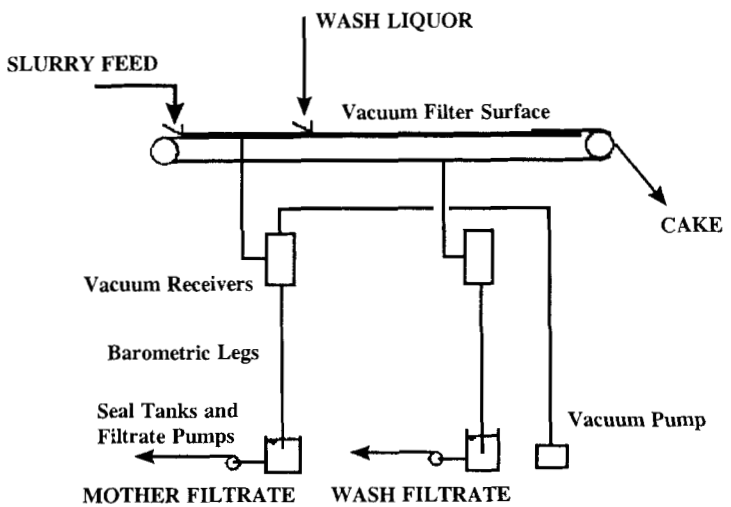

Figure 7. *Example of filtration flow scheme.*

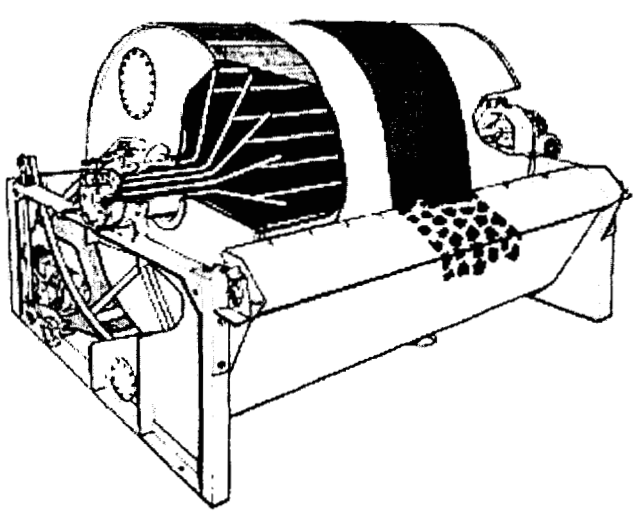

Figure 8. *Rotary drum filter.*

The key features of the a rotary drum filter are as follows. First, the machine has a drum that is supported by a large diameter trunnion on the valve end and a bearing on the drive end, and its face is divided into circumferential sectors each forming a separate vacuum cell. The internal piping that is connected to each sector passes through the trunnion and ends up with a wear plate having openings which correspond to the number of sectors. The drum deck piping is arranged so that each sector has a leading pipe to collect the filtrate on the rising side of the drum and a trailing pipe to collect the remaining filtrate from the descending side to ensure complete evacuation prior to cake discharge. The drum is driven with a variable speed drive at speeds that normally range from 1 to 10 MPR. The machine is equipped with a valve with a bridge setting which controls the sequence of the cycle so that each sector is subjected to vacuum, blow and a dead zone. When a sector enters submergence, vacuum commences and continues to a point that it is cut-off and blow takes place to assist in discharging the cake. The valve has on certain filters adjustable blocks and on others a fixed bridge ring. Adjustable bridge blocks enable the optimization of form to dry ratio within the filtration cycle as well as the submergence of the drum in the tank. A cake discharge mechanism is available in several forms, such as a scraper, belt, roll and in very rare cases a string discharge. Blow is applied only to filters with scraper and roll discharge mechanisms but not to filters with a belt or string discharge. The various types of cake discharge mechanisms are illustrated in Figures 9 through 12.

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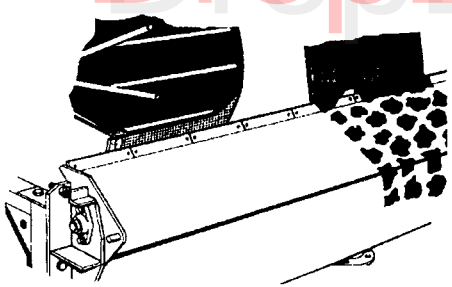

Figure 9. *Shows scraper discharge.*

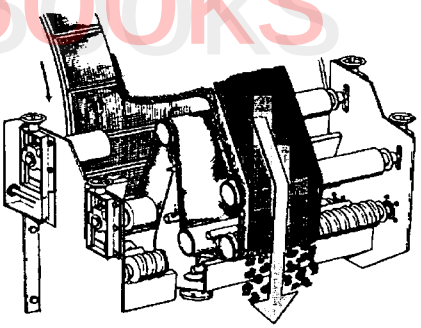

Figure 10. *Shows a belt discharge.*

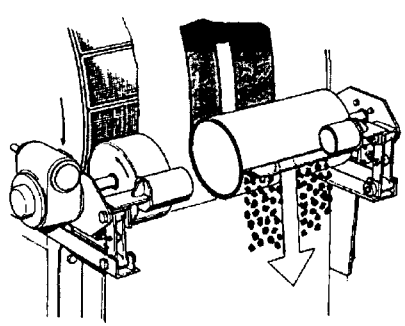

Figure 11. *Shows a roll discharge.*

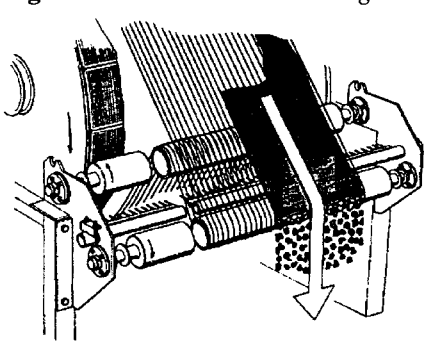

Figure 12.. *Shows a string discharge.*

A mild agitator is used to keep the slurry in suspension and reciprocates between the drum face and tank bottom at 16 or so CPM. The system is equipped with a tank with baffled slurry feed connections, an adjustable overflow box to set a desired drum submergence and a drain connection. The tanks are normally designed for an apparent submergence of 33 to 35%, however on certain applications 50% and more is possible. With these special designs the tank ends are higher in order to accommodate stuffing boxes on both the drive shaft and valve end trunnion. On applications where cake washing is required, 2 or 3 manifolds with overlapping nozzles are mounted to a pair of splash guards that are bolted to the tank ends. The position of the manifolds and the quantity of wash liquid are adjustable depending on the wash characteristics of the cake. Optional controls may be used to automate settings such as drum speed, applied wash liquid and drum submergence for a desired cake thickness or throughput. The monitoring of drum submergence controls the slurry feed valves so an adjustable overflow weir is not necessary except for a fixed connection in case of emergency. In broad terms, drum filters are suitable to the following process requirements:

- Slurries with solids that do not tend to settle rapidly and will remain in a uniform suspension under gentle agitation,
- Cakes which do not require long drying times to reach asymptotic moisture values,
- Cakes when a single washing stage is sufficient to remove residual contaminants from the cake or yield maximum recovery of filtrate,
- Filtrates that generally do not require a sharp separation between the mother and wash filtrates. Some complex valves, however, enable atmospheric purging of the sectors and internal piping to facilitate a sharp separation of filtrates,
- Filtrates that are acceptable with a low quantity of fines that pass trough the filter cloth in the first few seconds of cake formation. Broadly, and depending on particle size and cloth permeability, the filtrate may contain 1,000 to 5,000 ppm insolubles.

The entire filtration cycle on a rotary drum filter must be completed within a geometry of 360 degrees. The cycle sequence of a single sector is illustrated in Figure 13, where it is assumed that the drum rotates in a clockwise direction. The stages of the cycles are cake formation, cake washing and drying, cake discharge, and the dead zone. Each is briefly described below.

Cake Formation: With the overflow weir set to a maximum the "apparent submergence" is normally 33 to 35% so the slurry levels between 0400 and 0800 hours. Once a sector is completely submerged vacuum is applied and a cake starts to form up to a point where the sector emerges from the slurry. The portion of the cycle available for formation is the "effective submergence" and its duration

depends on the number of sectors, the slurry level in the tank and the bridge setting which controls the form to dry ratio.

Cake Washing and Drying: After emerging from submergence the drying portion of the cycle commences and for non-wash applications continuous to about 0130 hours where the vacuum is cut-off. If cake washing is required the wash manifolds will be located from about 1030 to 1130 hrs and the remaining time to vacuum cut-off at 0130 is the portion allocated to final cake drying.

Cake Discharge: After vacuum for the entire sector is cut-off air blow commences at about 0200 hours in order to facilitate the discharge of the cake. The blow, depending on the position of the tip of the scraper blade, will cut-off at approximately 0300 hrs. Drum filters are normally operated with a low pressure blow but on certain applications a snap blow is applied and to avoid the snapping out of the caulking bars or ropes wire winding of the cloth is recommended . Blow is used on scraper and roll discharge mechanisms but on belt discharge filters vacuum cuts-off when the filter media leaves the drum.

Dead Zone: Once the blow is cut-off the sector passes through a zone blocked with bridges so that no air is drawn through the exposed filter media which might cause the loss of vacuum on the entire drum surface.

The slow rotation of the drum and reciprocation of the agitator reduce maintenance requirements to a minimum but the following should be inspected periodically. The strip liner of the trunnion bearing at the valve end will normally wear at the lower half. However, in situations when the slurry has a high specific gravity, the drum has a tendency to become buoyant causing wear to the upper half.

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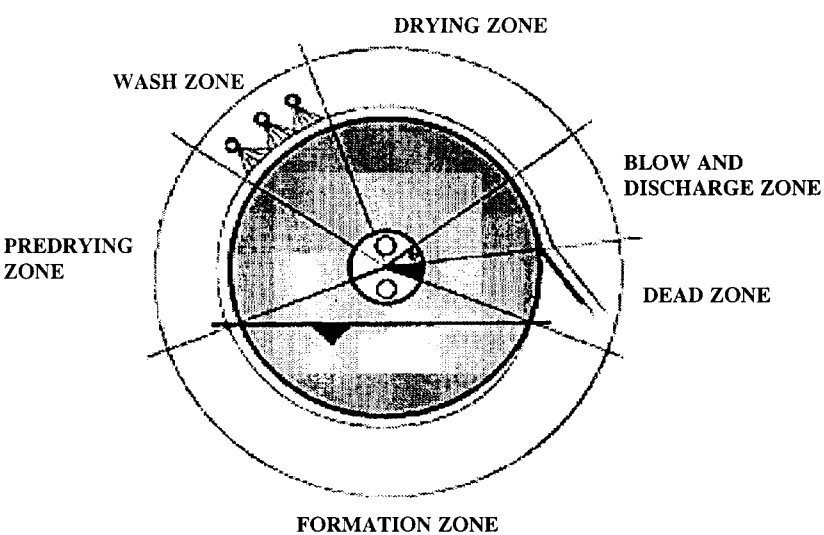

Figure 13. *Cycle sequence of a rotary drum filter.*

One way to remove the lower half of the liner, when hoisting facilities are not available or operational, is to float the drum by filling the tank with a sufficiently concentrated solution. Other components that should be periodically checked are:

- The stuffing boxes on high submergence filters should be inspected for leakage and, if necessary, the stud nuts should be tightened. It should be noted that excess tightening can increase substantially the load on the drum drive so the use of a torque wrench is recommended.
- The face of the wear plate should be checked periodically and remachined if necessary. A whistling noise during operation is an indication the wear plate is worn out or the valve spring requires tensioning.
- The drum has a bailer tube that protrudes from the drive end shaft and must be kept open at all times since its blockage may cause the collapse of the drum. The bailer tube is a tell-tale indication to the following: (a) if a lighter flame is drawn through the bailer tube to the inside of the drum it indicates that a vacuum leak exists in the drum shell or the internal piping; (b) if leakage is observed from the bailer tube it indicates that a hole exists in the drum head causing penetration of slurry from the tank into the drum.
- The on-line filter on the wash headers manifold should be checked periodically for pressure build- up due to progressive blockage. Likewise, the nozzles on the wash headers should be kept clean in order to ensure overlapping for full coverage of the washed cake.

Internal Rotary-Drum Filters: An example of an internal rotary-drum filter is illustrated in Figure 14. The filter medium is contained on the inner periphery. This design is ideal for rapidly settling slurries that do not require a high degree of washing. Tankless filters of this design consist of multiple-compartment drum vacuum filters.

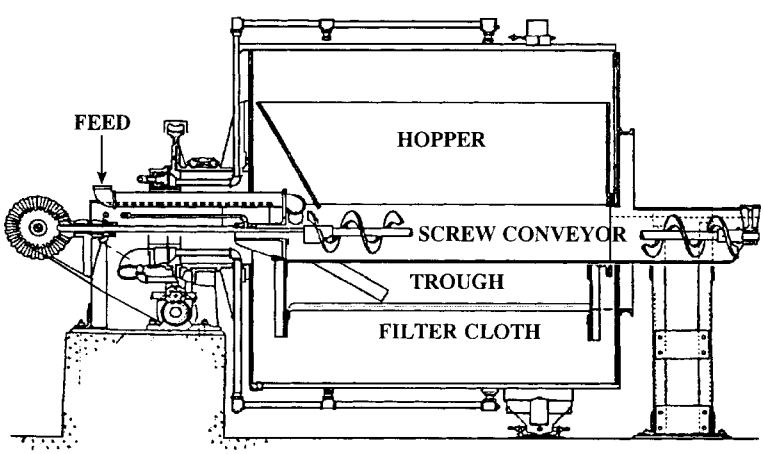

Figure 14. *Section view of interior medium rotary-drum vacuum filter.*

One end is closed and contains an automatic valve with pipe connections to individual compartments. The other end is open for the feed entrance. The drum is supported on a tire with rigid rollers to effect cake removal. The drum is driven by a motor and speed-reducer connected to a riding roll shaft. The feed slurry is discharged to the bottom of the inside of the drum from the distributor and is maintained as a pool by a baffle ring located around the open end and the closed portion of the outer end. As the drum revolves, the compartments successively pass through the slurry pool, where a vacuum is applied as each compartment becomes submerged. Slurry discharge is accomplished at the top center where the vacuum is cut off and gravity (usually assisted by blowback) allows the solids to drop off onto a trough. From there, a screw or belt conveyor removes the solids from the drum. This filter is capable of handling heavy, quick-settling materials.

Nutsche Filters: Nutsche filters are one design type with a flat filtering plate. This configuration basically consists of a large false-bottomed tank with a loose filter medium. Older designs employ sand or other loose, inert materials as the filtering medium, and are widely employed in water clarification operations. In vacuum filtration, these false-bottom tanks are of the same general design as the vessels employed for gravity filtration. They are, however, less widely used, being confined for the most part to rather small units, particularly for acid work. Greater strength and more careful construction are necessary to withstand the higher pressure differentials of vacuum over gravity. This naturally increases construction costs. However, when high filtering capacity or rapid handling is required with the use of vacuum, the advantages may more than offset higher costs. Construction of the vacuum false-bottom tank is relatively simple; a single vessel is divided into two chambers by a perforated section. The upper chamber operates under atmospheric pressure and retains the unfiltered slurry. The perforated false bottom supports the filter medium. The lower chamber is designed for negative pressure, and to hold the filtrate. Nutsche filters are capable of providing frequent and uniform washings. A type of continuous filter that essentially consists of a series of Nutsche filters is the rotating-tray horizontal filter.

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Horizontal Rotary Filters: An example of a horizontal rotary filter is illustrated in Figure 15. These machines are well suited to filtering quick-draining crystalline-like solids. Due to its horizontal surface, solids are prevented from falling off or from being washed off by the wash water. As such, an unusually heavy layer of solids can be tolerated. The basic design consists of a circular horizontal table that rotates about a center axis. The table is comprised of a number of hollow pie-shaped segments with perforated or woven metal tops. Each of the sections is covered with a suitable filter medium and is connected to a central valve mechanism that appropriately times the removal of filtrate and wash liquids and the dewatering of the cake during each revolution. Each segment receives the slurry in succession. The wash liquor is sprayed onto each section in two applications. Then the cake is dewatered by passing dry air over it. The cake is finally removed by a scraper.

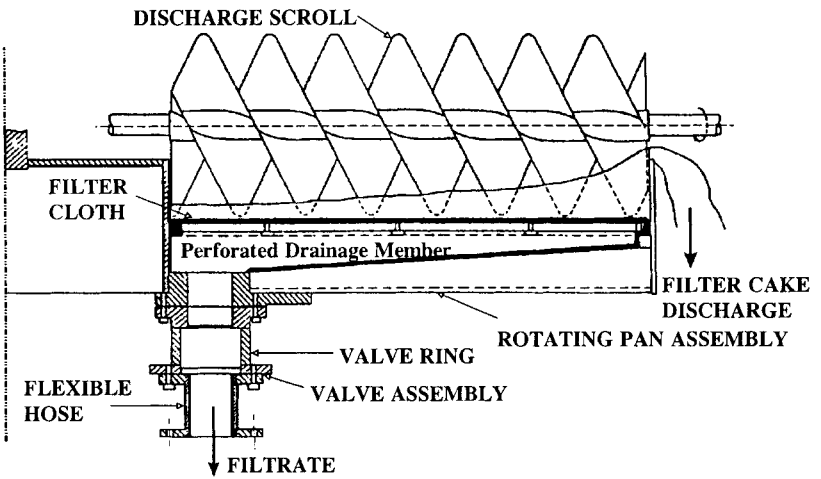

Figure 15. *Cross section of a rotary horizontal vacuum filter.*

Belt Filters: Belt filters consist of a series of Nutsche filters moving along a closed path. Nutsche filters are connected as a long chain so that the longitudinal edge of each unit has the shape of a baffle plate overlapping the edge of the neighboring unit. Each unit is displaced by driving and tensioning drums. Nutsche filters are equipped with supporting perforated partitions covered with the filtering cloth. The washed cake is removed by turning each unit over. Sometimes a shaker mechanism is included to ensure more complete cake removal. A belt filter can be thought of as consisting of an endless supporting perforated rubber belt covered with the filtering cloth. The basic design is illustrated in Figure 16. Supporting and filtering partitions are displaced by driving drum and maintained in a stretched condition by tensioning the drum, which rotates due to friction against the rubber belt. Belt edges (at the upper part of their path) slide over two parallel horizontal guide planks. The elongated chamber is located between the guide planks. The chamber in the upper part has grids with flanges adjoining the lower surface of the rubber belt. The region under the belt is connected by nozzles to the filtrate collector, which is attached to a vacuum source. The chamber and collector are divided into sections from which filtrate and washing liquid may be discharged. The sludge is fed by the trough. The cake is removed from the drum by gravity or blowing air, or sometimes it is washed off by liquid from the distributor nozzle. The washing liquid is supplied from a dike tank, which can move along the filtering partition. It can be washed during the belt's motion along the lower path. The filtering partition, illustrated in Figure 17, consists of a riffled rubber belt with slots, grooves and the filter cloth, which is fixed in a set of grooves by cords. Slots through which the filtrate passes are located over the grids of the elongated chamber. The edges of the rubber belt are bent upward by guides forming a gutter on the upper path of the

belt. The velocity of the filtering partition depends on the physical properties of the sludge and the filter length. The cake thickness may range from 1 to 25 turns. The advantages of belt filters are their simplicity in design compared to filters with automatic valves, and the abilities to provide countercurrent cake washing and removal of thin layers of cake. Their disadvantages include large area requirements, inefficient use of the total available filter area, and poor washing at the belt edges.

Cross Mode Filters: Filters of this group have a vertical flat or cylindrical filtering partition. In this case, filtrate may move inside the channels of the filtering elements along the surface of the filtering partition downward under gravity force action, or rise along this partition upward under the action of a pressure differential. In the separation of heterogeneous suspensions, nonuniform cake formation along the height can occur because larger particles tend to settle out first. This often results in poor cake washing due to different specific resistance over the partition height.

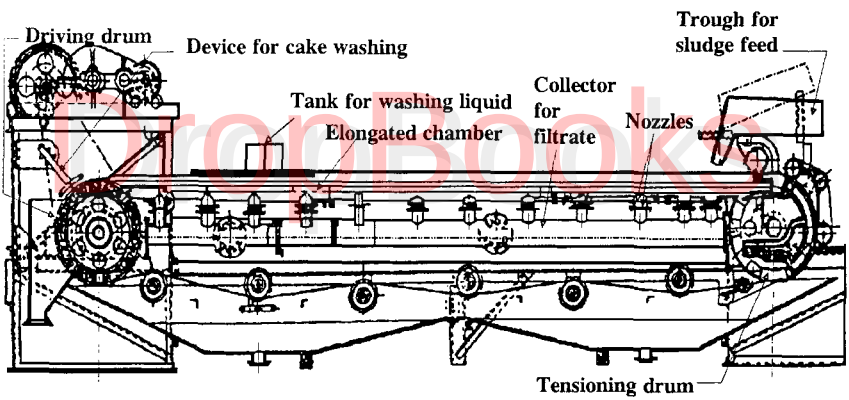

Figure 16. *Basic design features of a belt filter.*

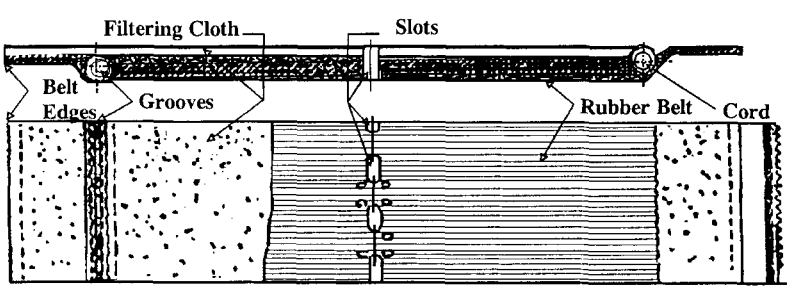

Figure 17. *Filtering partition for a belt filter.*

The cake may creep down along the partition due to gravity; this is almost inevitable in the absence of a pressure gradient across the filtering partition. The vertical filtering partition makes these filters especially useful as thickeners, since it is convenient to remove cake by reverse filtrate flow.

Filter Presses: The most common filter press is the plate-and-frame design, consisting of a metal frame made up of two end supports rigidly held together by two horizontal steel bars. Varying numbers of flat plates containing cloth filter media are positioned on these bars. The number of plates depends on the desired capacity and cake thickness. The plates are clamped together so that their frames are flush against each other, forming a series of hollow chambers. The faces of the plates are grooved, either pyramided or ribbed. The entire plate is covered with cloth, which forms the filtering surface. The filter cloth has holes that register with the connections on the plates and frames, so that when the press is assembled these openings form a continuous channel over the entire length of the press and register with the corresponding connections on the fixed head. The channel opens only into the interior of the frames and has no openings on the plates. At the bottom of the plates, holes are cored so that they connect the faces of the plates to the outlet cocks. As the filterable slurry is pumped through the feed channel, it first fills all of the frames. As the feed pump continues to supply fluid and build up pressure, the filtrate passes through the cloth, runs down the face of the plate and passes out through the discharge cock. When the press is full, it is opened and dumped. The cake cannot be washed in these units and is therefore discharged containing a certain amount of filtrate with whatever valuable or undesirable material it may contain. Each plate discharges a visible stream of filtrate into the collecting launder. Hence, if any cloth breaks or runs cloudy, that plate can be shut off without spoiling the entire batch. If the solids are to be recovered, the cake is usually washed. In this case, the filter has a separate wash feed line. The plates consist of washing and nonwashing types arranged alternately, starting with the head plate as the first nonwashing plate. The wash liquor moves down the channels along the side of each washing plate, and moves across the filter cake to the opposite plate and drains toward the outlet. This is illustrated in Figure 18. To simplify assembly, the nonwashing plates are marked with one button and the washing plates with three buttons. The frames carry two buttons. In open-delivery filters the cocks on the one-button plates remain open and those on the three-button plates are closed. In closed-delivery filters a separate wash outlet conduit is provided. Figure 19 illustrates the basic design of a frame, a nonwashing plate and a washing plate. Initial capital investment and floor space requirements are low in comparison to other types of filters. They can be operated at fall capacity (i.e., all frames in use) or at reduced capacity by blanking off some of the frames by dummy or blank plates. They can deliver reasonably well washed and relatively dry cakes. However, the combination of labor charges for removing the cakes and fixed charges for downtime may constitute a high percentage of the total cost per operating cycle.

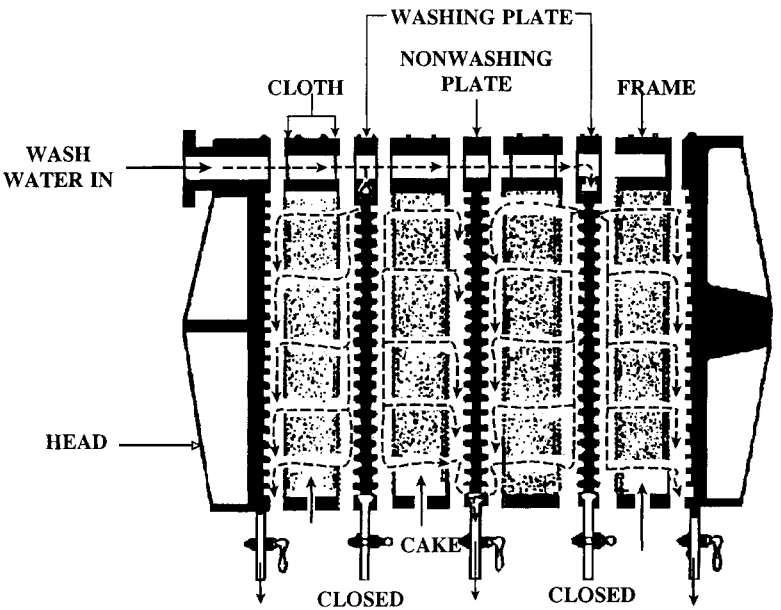

Figure 18. *Wash water outlets on a filter press.*

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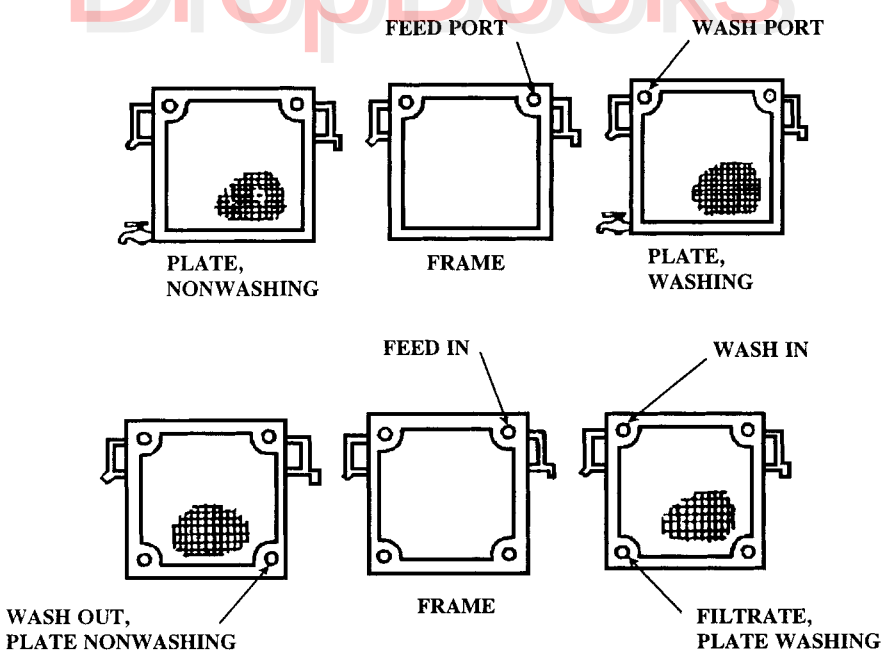

Figure 19. *Plate and frames for open and closed delivery systems.*

Leaf Filters: Leaf filters are similar to plate-and-frame filters in that a cake is deposited on each side of a leaf and the filtrate flows to the outlet in the channels provided by a coarse drainage screen in the leaf between the cakes. Figure 20 illustrates the basic scheme. The leaves are immersed in the sludge when filtering, and in the wash liquid when washing. Therefore, the leaf assembly may be enclosed in a shell, as in pressure filtration, or simply immersed in sludge contained in an open tank, as in vacuum, filtration. When operating a pressure leaf filter, the sludge is fed under pressure from the bottom and is equally distributed. The clear filtrate from each leaf is collected in a common manifold and carried away. In filters with an external filtrate manifold as illustrated in Figure 21, the filtrate from each leaf is visible through a respective sight glass. This is not possible when the leaves are mounted on a hollow shaft that serves as an internal filtrate collecting manifold. The filter cakes build on each side of the leaves as filtration continues, until a specified cake thickness is achieved. During the cake washing phase, excess sludge is first drained while compressed air is introduced at about 3 to 5 psi. The role of the compressed air is to prevent the cake from pealing off of the leaves during the washing stage.

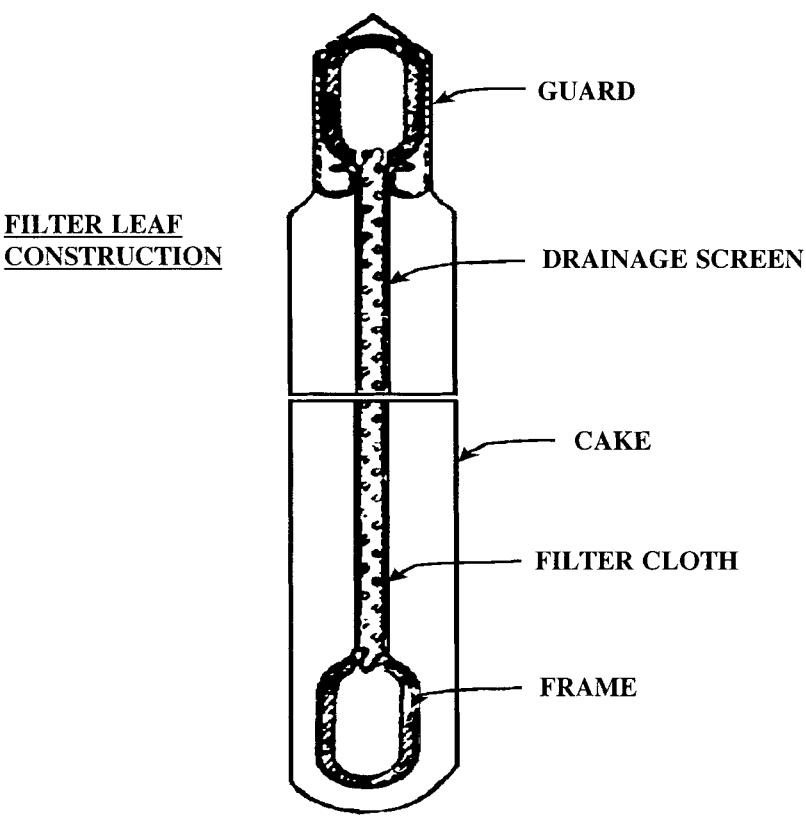

Figure 20. *Sectional view.*

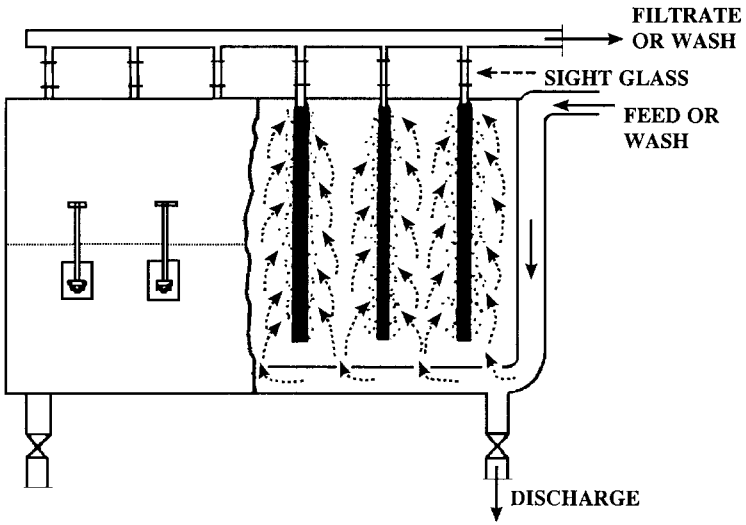

Figure 21. *Sweetland pressure filter.*

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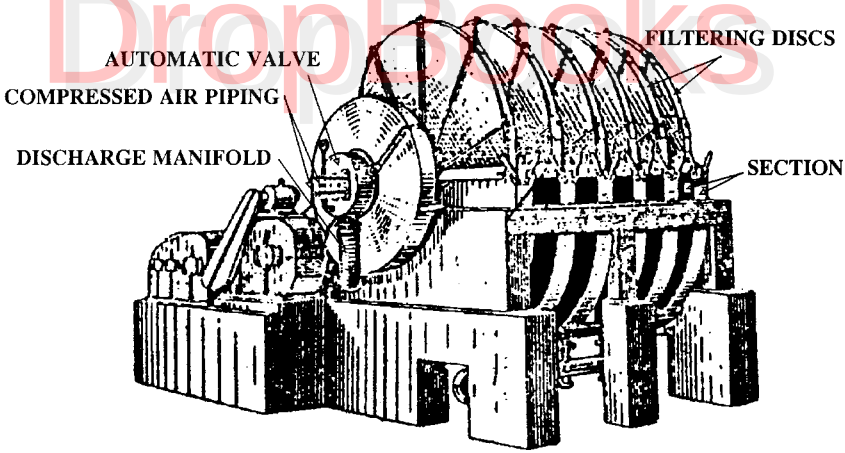

Figure 22. *Rotary-disc vacuum filter.*

Disk Filters: Disk filters consist of a number of concentric disks mounted on a horizontal rotary shaft. The operating principle is the same as that of rotary-drum vacuum filters. The basic design is illustrated in Figure 22. The disks are formed by using V-shaped hollow sectors assembled radially about a central shaft. Each sector is covered with filter cloth and has an outlet nipple connected to a manifold

extending along the length of the shaft and leading to a port on the filter valve. Each row of sectors is connected to a separate manifold. The sludge level in the tank should provide complete submergence to the lowest sector of the disks. Compared to drum vacuum filters, disk filters require considerably less floor space for the same filter area. However, because these machines have vertical filtering surfaces, cake washing tends not to be as efficient as in the case of a drum filter. The disk filter is best suited for cases when the cake does not require washing and floor space is critical.

Cartridge Filters: Cartridge filters are used in a multitude of solid-liquid filtration applications ranging from laboratory scale operations to industrial flows in excess of 5,000 gpm. These units are typically operated in the countercurrent mode. Common configurations consist of a series of thin metal disks that are 3 to 10 inches in diameter, set in a vertical stack with very narrow uniform spaces between them. The disks are supported on a vertical hollow shaft, and fit into a closed cylindrical casing. Liquid is fed to the casing under pressure, whence it flows inward between the disks to openings in the central shaft and out through the top of the casing. Solid particles are captured between the disks and remain on the filter media. Since most of the solids are removed at the periphery of the disks, the unit is referred to as an edge filter. The accumulated solids are periodically removed from the cartridge. As with any filter, careful media selection is critical. Media that are too coarse, for example, will not provide the needed protection. However, specifying finer media than necessary can add substantially to both equipment and operating costs. Factors to be considered in media selection include the solids loading, the nature and properties of the particles, particle size, shape and size distribution, the amount of solids to be filtered, fluid viscosity, slurry corrosiveness, abrasiveness, adhesive qualities, liquid temperature, and flowrate. Typical filter media are wire mesh (typically 10 to 700 mesh), fabric (30 mesh - 1 μ), slotted screens (10 mesh to 25 μ) and perforated stainless steel screens (10 to 30 mesh). Multiple filters are also common, consisting of two or more single filter units valved in parallel to common headers. The distinguishing feature of these filters is the ability to sequentially backwash each unit in place while the others remain on stream. Hence, these systems are continuous filters. These units can be fully automated to eliminate manual backwashing. Backwashing can be controlled by changes in differential pressure between the inlet and outlet headers. One possible arrangement consists of a controller and solenoid valves that supply air signals to pneumatic valve actuators on each individual filter unit. As solids collect on the filter elements, flow resistance increases. This increases the pressure differential across the elements and, thus, between the inlet and outlet headers on the system. When the pressure drop reaches a preset level, an adjustable differential pressure switch relays information through a programmer to a set of solenoid valves, which in turn sends a signal to the valve actuator. This rotates the necessary valve(s) to backwash the first filter element. When the first element is cleaned and back on stream, each successive filter element

is backwashed in sequence until they are all cleaned. The programmer is then automatically reset until the rising differential pressure again initiates the backwashing cycle. Filter cartridges or tubes are made from a variety of materials. Common designs are natural or synthetic fiber wound over a perforated plastic or metal core. A precision winding pattern covers the entire depth of the filter tube with hundreds of funnel-shaped tunnels, which become gradually finer from the outer surface to the center of the tube and trap progressively finer particles as the fluid travels to the center. This provides greater solids retention capacity than is associated with surface filter media of the same dimensions. Typical cartridge materials are cotton, Dynel, polypropylene, acetate, porous stone and porous carbon filter lubes. Supporting perforated cores for cotton, Dynel or polypropylene are stainless steel, polypropylene or steel. Supporting cores for acetate tubes are tin-plated copper with voile liner. Porous stone and porous carbon filter tubes do not require supporting cores. Stainless steel cores are recommended for mildly acid and all alkaline solutions, i.e., pH 4-14. Polypropylene cores are used where all metal contact must be eliminated or where stainless steel is attacked, such as high chloride and sulfuric acid solutions. Two types of polypropylene cores are commonly used: mesh polypropylene and rigid perforated polypropylene. Mesh polypropylene is satisfactory for temperatures below 140 °F. The more expensive rigid polypropylene cores are used for temperature applications over 140 °F, and for double- and triple-tiered filter chambers because greater strength is needed here. Perforated steel cores are used for dilute alkaline solutions, solvents, lacquers, oils, emulsions, etc.

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Strainers and Filter Baskets: Strainer filter baskets and filter bag baskets are used as prefiltering devices. This prestraining or prefiltering prevents larger contaminated particles from continuing through the filtration process and thus extends the life of the entire system. Single-stage strainers and bag filters differ only in the basket design. Strainer baskets have solid flat bottoms, and baskets for filter bags have perforated bottoms to accept standard size filter bags. Dual-stage straining/ filtering action is achieved by insertion of a second, inner basket. It is supported on the top flange of the outer basket. Both baskets can be strainers (with or without wire mesh linings) or both can be baskets for filter bags. They may also be a combination: one a strainer basket, the other a filter bag basket. Dual-stage action increases strainer or filter life and reduces servicing needs. All baskets are equipped with a seal. The seal is maintained during operation by a hinged basket bail handle being held down under the closed cover, which holds the basket down against a positive stop in the housing. Figure 23 illustrates the design features. Fabric bag filter baskets are capable of providing removal ratings from 20 mesh to nominal 1 micron (μ) for both Newtonian and viscous liquids. Wire mesh or fabric baskets can be cleaned and reused in many applications, or are disposable when cleaning is not feasible. Side-entry models feature permanent flanged connections, for line pressures to 150 psi. These filters are fabricated to American Society of Mechanical Engineers (ASME) codes for applications that must comply with piping

standards established in many processing plants. Top-entry models feature the inlet connection as an integral part of the lid. The inlet can be equipped with different types of quick disconnects for fast basket removal. Strainers should be selected so that the pressure drop incurred does not exceed a specified limit with a clean strainer basket (typically 2 psi). Pressure drop versus flow capacity curves for basket strainers are given in Figure 24. This plot provides gross pressure drop for different capacities of water flow at suitable strainer pipe sizes. The value obtained must be corrected on the basis of the actual fluid viscosity and strainer opening size to be used. These corrections are given in table accompanying the plot in Figure 24, using the following stepwise procedure - Step (1): Under the pressure drop value from the bottom scale in Figure 24, with the specified flowrate - read up to where its vertical line intersects the diagonal representing a strainer pipe size that gives a reasonable pressure drop, which is found by following the horizontal line to the pressure drop scale at the left. Step (2): To correct this figure to match the actual fluid viscosity (and strainer media selection), use the table values with Figure 24. Read down the appropriate viscosity column and across from the appropriate strainer media description to obtain the correction factor. Step (3): Multiply the pressure drop figure obtained under step 2 by the correction factor to obtain the adjusted pressure drop. Step (4): If you are using a mesh-lined (not pleated) strainer basket, the pressure drop can be lowered by using a 30-in.-deep basket instead of a 15-in.-deep basket; divide the pressure drop figure obtained from step 3 by 1.5.

Diaphragm Filters: Diaphragm filters are special case of filter presses. That have the ability to reduce sludge dewatering costs by a squeezing cycle using a diaphragm. Instead of the conventional plate-and-frame unit in which constant pumping pressure is used to force the filtrate through the cloth, diaphragm filters combine an initial pumping followed by a squeezing cycle that can reduce the process cycle time by as much as 80%. The operating cycles for this design are illustrated in Figure 25.

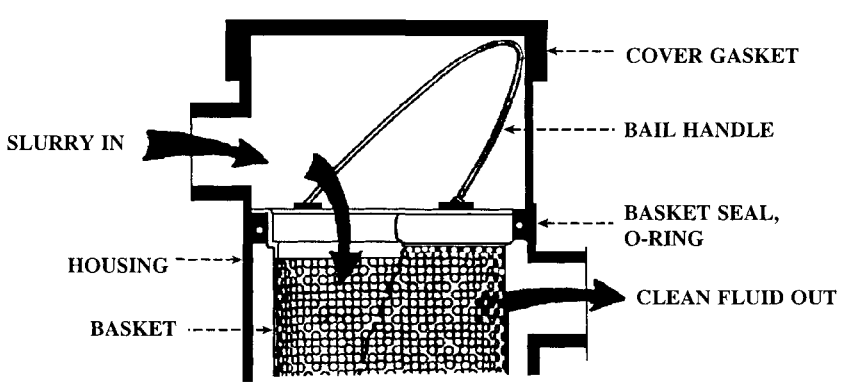
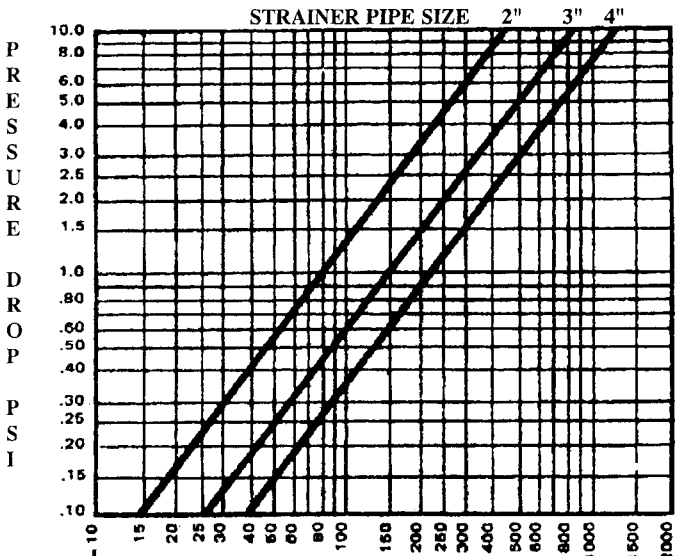


Figure 23. *Upper portion of basket with seal.*



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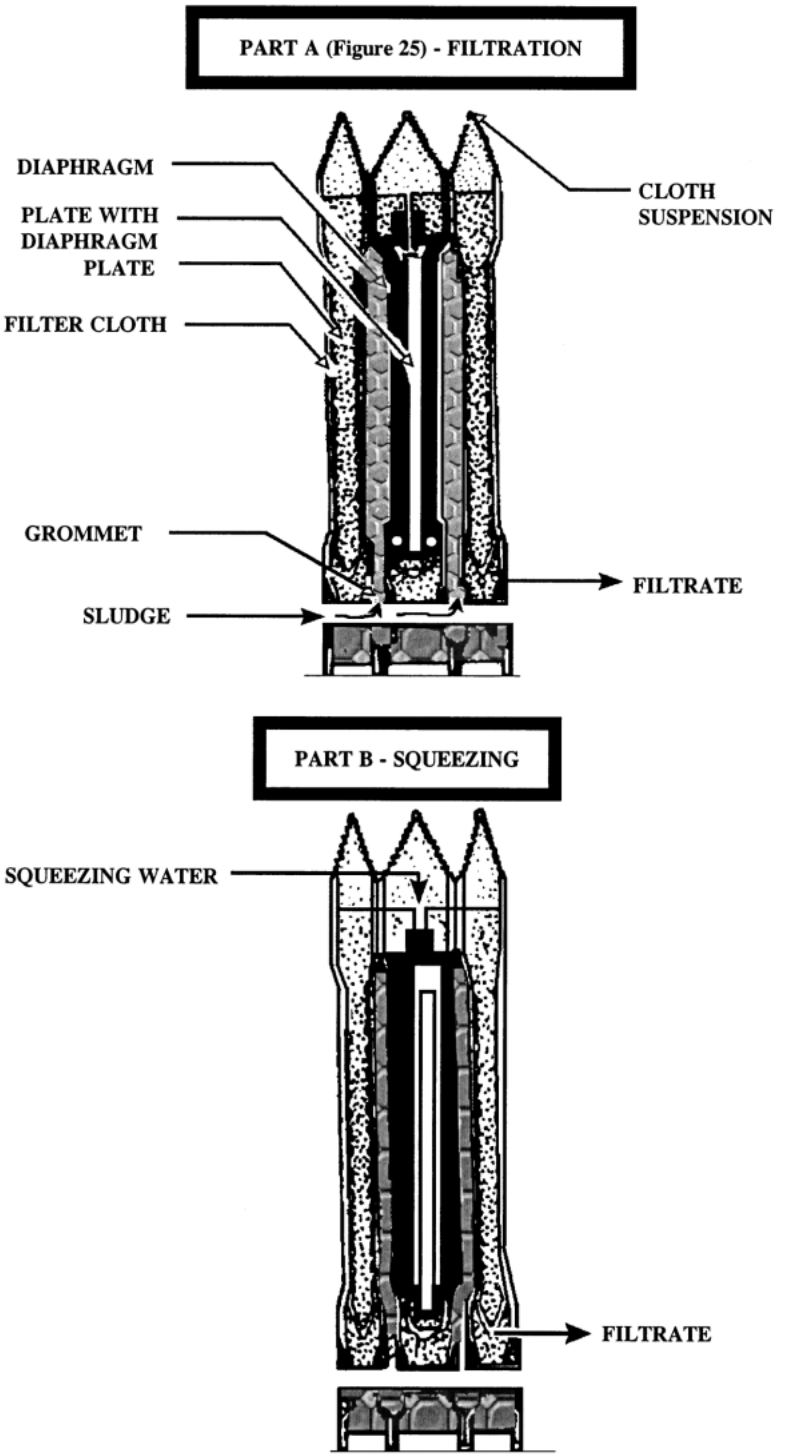
VISCOSITY CORRECTION FACTORS

	Viscosity (cP)								
	10	50	100	200	400	600	800	1000	2000
Air unlined baskets, with or without picated inserts	0.65	0.85	1.00	1.10	1.20	1.40	1.50	1.60	1.80
40-mesh lined	0.73	0.95	1.20	1.40	1.50	1.80	1.90	2.00	2.30
60-mesh lined	0.77	1.00	1.30	1.60	1.70	2.10	2.20	2.30	2.80
80-mesh lined	0.93	1.20	1.50	1.90	2.10	2.40	2.60	2.80	3.50
100-mesh lined	1.00	1.30	1.60	2.20	2.40	2.70	3.00	3.30	4.40

Figure 24. *Pressure drop versus flow rate relation fro basket strainers.*

During the filtration cycle, sludge is fed at approximately 100 psi into each chamber through an inlet pipe in the bottom portion of the filter plate. The number of chambers can range from a few dozen to more than 100. The sludge feed pump continues to feed sludge into the chamber until a predetermined filtering time has been achieved. Filtrate passes onto the cloth on both sides of the chamber. The filtration cycle is completed independently in each chamber. Short filtration cycles produce cake thicknesses of 0.5 to 0.75 in. (12.7-19.1 mm). Once the filtration cycle is complete, the sludge pump is stopped and a diaphragm in the chamber is expanded by water pressurized up to 250 psi. This compresses the sludge on both sides of the chamber into a thin, uniform cake with a solids content of more than 35%.


PART A (Figure 25) - FILTRATION
DIAPHRAGM
PLATE WITH DIAPHRAGM PLATE
FILTER CLOTH
CLOTH SUSPENSION
GROMMET
FILTRATE
SLUDGE
PART B - SQUEEZING
SQUEEZING WATER
FILTRATE

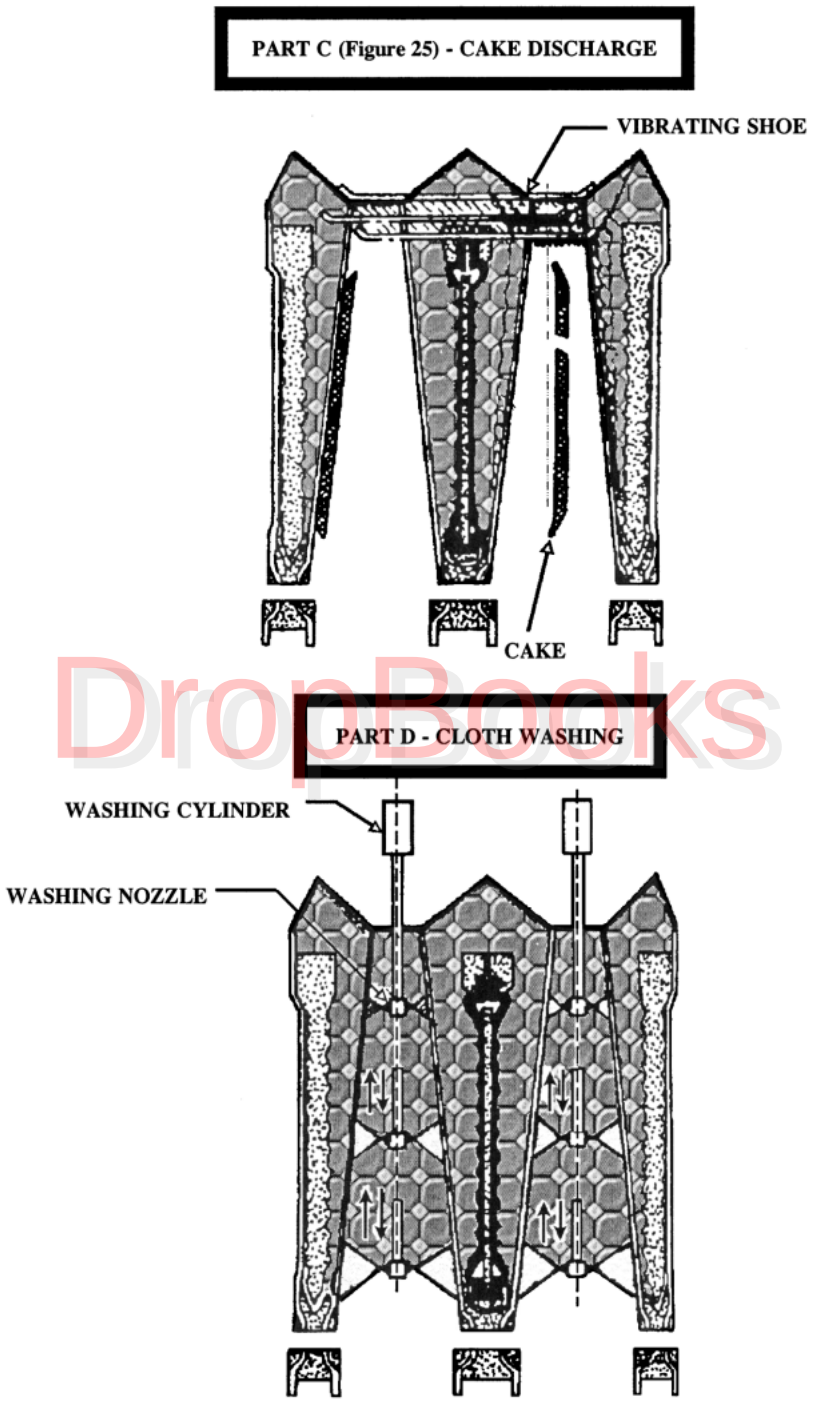


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Figure 25. *Diaphragm press operating cycles.*

The uniform water content of the thin cake (no wet cores) results in easier shredding and conveying and makes it much more adaptable to self-sustained thermal destruction or landfill. Optimum filtering and squeezing time cycles vary, depending on the type of sludge, and can be determined accurately by bench tests. The squeezing water is recycled. A hydraulic ram keeps the chambers in position during both cycles. On completion of the filtration and squeeze cycles, the chambers are automatically opened and the cakes are discharged, usually onto a belt conveyor. No precoating is required. Two chambers are normally opened at a time in sequence. This reduces the impact loading on the belt conveyor. Any sludge or filtrate remaining in the feed and filtrate lines is automatically purged by high-pressure (100-psi) air before the next cycle begins. The purging prevents wet sludge from discharging and keeps sludge lines from plugging.Cake discharge from filter presses is a relatively fast step. After a number of cycles (depending on the sludge type), the filter cloth will require cleaning. This can be accomplished manually or can be performed automatically at preset frequencies with an automatic cloth washer using a jet of 1,000-psi wash water. Where even faster cake discharge is desired or where sludge cakes may tend to be sticky, automatic cloth vibrators can be provided. These units help speed mechanical discharge and help remove cakes where poor sludge conditioning causes excessive sticking. This reduces the need for continuous monitoring by operators. Cloth vibrators also simplify cloth selection, since cloths can be selected to assure clearer filtrate or better filtering qualities rather than sacrificing these advantages for a cloth that allows for filter discharge characteristics. The process of cake discharging is illustrated in Figure 26.

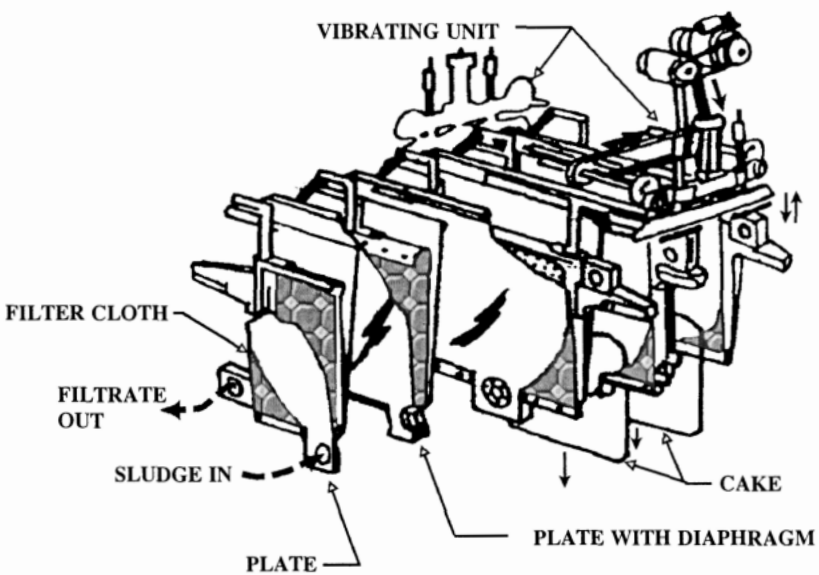

Figure 26. *Cake discharge from a diaphragm filter press.*

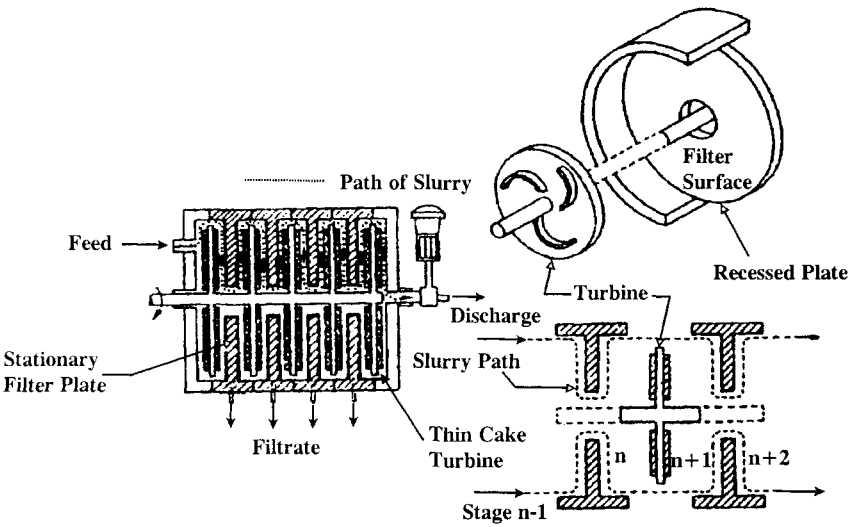

Figure 27. *Components of a staged, thin-film filter/thickener.*

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High Pressure, Thin Cake Filters: Thin-cake staged filters have been used effectively at high flowrates per unit area for many years. Use of ultra-thin cakes is a useful technique for increasing flowrates. The basic elements of the filter are illustrated in Figure 27. Filter surfaces are recessed plates equipped with rotating turbines that maintain permanent precoat-type thin cakes throughout the filter. Cake thickness is prevented from growing beyond the in situ precoat formed during the first few minutes of the operation by blades on a rotating shaft passing through the axis of the filter. Slurry flows into the first stage and then flows around the turbines and through the clearances between the shaft and active filter surfaces. As liquid is removed, the thickening slurry moves from stage to stage. The unit acts as a filter-thickener, producing a continuous discharge that may contain a higher solids content than is normally encountered in conventional filters. The turbine plates sweep close to the filter cloths, leaving a thin, permanent cake on each stationary plate. Even in the last stage of the filter, where the slurry is highly non-Newtonian, a thin, easily identifiable, hard cake is maintained. At low turbine velocities, the blades serve as scrapers that limit the cake thickness to be dimensions of the clearance. At higher velocities, the cake thickness is reduced and can be as thin as 1.0 mm with 3-mm clearance. For filters that depend entirely on fluid action, shear forces at the cake surface depend on fluid properties and velocity distributions. The combination of high pressure (300 psi) and thin cakes produces high rates. Washing is accomplished either cocurrently or countercurrently. Separate filters in series can be employed in a manner similar to conventional thickeners. Washing may also be

performed within a single unit, whereby, an initial portion of the filter must be used to remove liquid. In this case, the final stages are used for concentration. Clean wash liquid may be injected after the initial filtering at one or several intermediate stations. Injection wash tends to increase the overall filtrate rate but decreases the cake output rate.

Centrifugal Filtration: Filtering centrifuges are distinguished from standard centrifugation by a filtering medium incorporated into the design. Slurry is fed to a rotating basket or bowl having a slotted or perforated wall covered with a filtering medium such as canvas or metal- reinforced cloth. The angular acceleration produces a pressure that transports the liquor through the filtering medium, leaving the solids deposited on the filter medium surface as a cake. When the feed stream is stopped and the cake spun for a short time, residual liquid retained by the solids drains off. This results in final solids that are considerably drier than those obtained from a filter press or vacuum filter. Principal types of filtering centrifuges are suspended batch machines, automatic short-cycle batch machines and continuous conveyor centrifuges. In suspended centrifuges, the filter medium is usually canvas or a similar fabric, or woven metal cloth. Automatic machines employ fine metal screens. The filter medium in conveyor centrifuges is usually the slotted wall of the bowl itself. Figure 28 provides an example of this machine. The system combines the features of a centrifuge and a screen. Feed enters the unit at the top and is immediately brought up to speed and distributed outward to the screen surface by a set of vanes. Water or other liquid is forced by the sudden centrifugal action through the screen openings into an effluent housing.

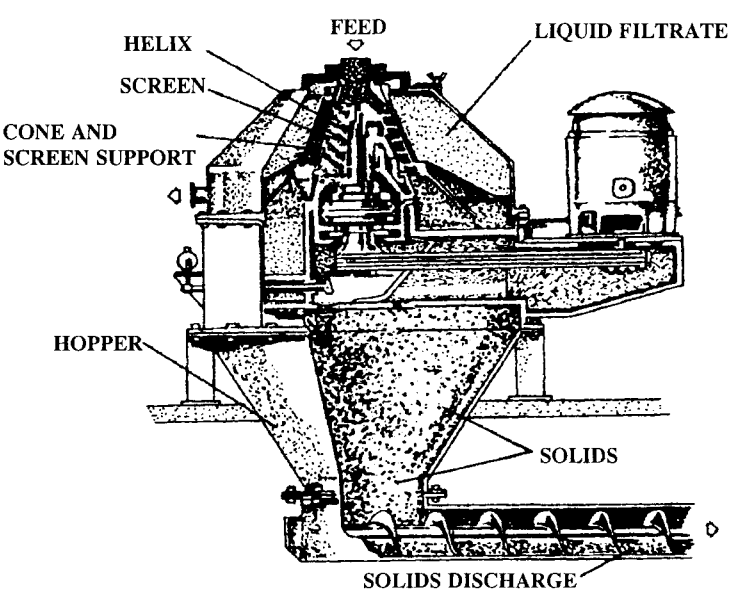

Figure 28. *Cutaway view of filter centrifuge.*

As solids accumulate, they are gently moved down the screen by the slightly faster rotating helix. With the increase in screen diameter, higher centrifugal gravities are encountered and solids are dispersed over a gradually increasing area, thus forming a thin, compact cake from which the remaining liquid is extracted. The relatively dry solids are blown out the bottom of the rotor by a set of vanes into a conical collection hopper.

Rapid Filteration: Rapid filtration is a standard unit operation in wastewater treatment. The operation is performed either in open gravitational flow filters or in closed pressure filters. Rapid pressure filters have the advantage of being able to be inserted in the pumping system, thus allowing use of a higher effective loadings. Note that pressure filters are not subject to development of negative pressure in a lower layer of the filter. These filters generally support higher speeds, as the available pressure allows a more rapid flow through the porous medium made up by the filter sand. Pressure filtration is generally less efficient than the rapid open type with free-flow filtration. Pressure filters have the following disadvantages. The injection of reagents is complicated, and it is more complicated to check the efficiency of backwashing. Work on the filter mass is difficult considering the assembly and disassembly required. Also, the risk of breakthrough by suction increases. Another disadvantages is that pressure filters need a longer filtration cycle, due to a high loss of head available to overcome clogging of the filter bed. Another option is to use open filters, which are generally constructed in concrete. They are normally rectangular in configuration. The filter mass is posed on a filter bottom, provided with its own drainage system, including bores that are needed for the flow of filtered water as well as for countercurrent washing with water or air. There are several types of washing bottoms. One type consists of porous plates which directly support the filter sand, generally without a layer of support gravel. Even if the system has the advantage of being of simple construction, it nevertheless suffers from incrustation. This is the case for softened water or water containing manganese. Porous filters bottoms are also subject to erosion or disintegration upon the filtration of aggressive water. The filter bottom is often comprised of pipes provided with perforations that are turned toward the underpart of the filter bottom and embedded in gravel. The lower layers are made up of gravel of approximate diameter 35 - 40 mm, decreasing up to 3 mm. The filter sand layer, located above this gravel layer, serves as a support and equalization zone. Several systems of filter bottoms comprise perforated self-supporting bottoms or false bottoms laid on a supporting basement layer. The former constitutes a series of glazed tiles, which includes bores above which are a series of gravels in successive layers. All these systems are surpassed to some extent by filter bottoms in concrete provided with strainers. The choice of strainers should in part be based on the dimensions of the slits that make it possible to stop the filter sand, which is selected as a function of the filtration goal. Obstruction or clogging occurs only rarely and strainers are sometimes used. Strainers may be of the type with an end that continues under the

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filter bottom. These do promote the formation of an air space for backwashing with air. If this air space is not formed, it can be replaced by a system of pipes that provide for an equal distribution of the washing fluids. Pressure filters are worth noting. These are usually set up in the form of steel cylinders positioned vertically. Another variation consists of using horizontal filtration groups. This has the drawback that the surface loading is variable in the different layers of the filter bed; moreover, it increases with greater penetration in the filter bed (the infiltration velocity is lowest at the level of the horizontal diameter of the cylinder). The filter bottom usually consists of a number of screens or mesh sieves that decrease in size from top to bottom or, as an alternative, perforated plates supporting gravel similar to that used in the filter bottoms of an open filter system. Filter mass washing can influence the quality of water being filtered. Changes may be consequent to fermentation, agglomeration, or formation of preferential channels liable to occur if backwashing is inadequate. Backwashing requires locating a source that will supply the necessary flow and pressure of wash water. This water can be provided either by a reservoir at a higher location or by a pumping station that pumps treated water. Sometimes an automated system is employed with washing by priming of a partial siphon pumping out the treated water stored in the filter itself. An example is shown in Figure 29. The wash water must have sufficient pressure to assure the necessary flow.

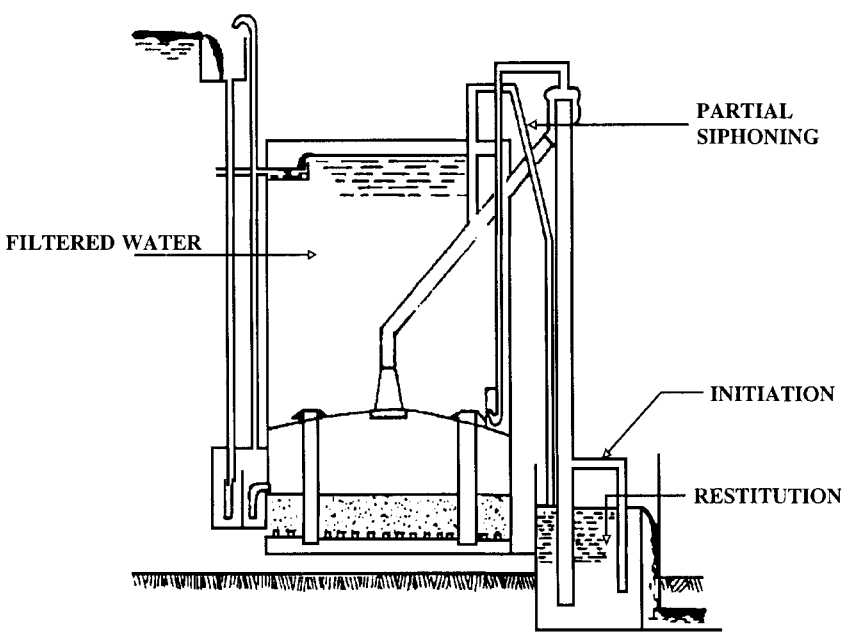

Figure 29. *Automatic backwashing filter with a partial siphon.*

Washing of the filter sands is accomplished followed by washing with water and in most cases including a short intermediate phase of simultaneous washing with air and water. The formation of a superficial crust on the filter sand is avoided by washing with air. After washing with air, water flow is gradually superimposed on the air flow. This operational phase ends at the same time that the wash air is terminated, to avoid the filter mass being blown away. The wash water contains materials that eventually require treatment in a sludge treatment plant. Their concentration varies as a function of the washing cycle. Accounting for the superficial load in filtration, velocity of the wash water, and length of the filtration cycle, it may be assumed that the water used for washing will not attain 5 % of the total production. For new installations the first washing cycles result in the removal of fine sand as well as all the other materials usually undesirable in the filter mass, such as particles of bitumen on the inner surface of the water inlet or other residuals from the crushing or straining devices of the filter media. Consequently, it is normal that at the beginning of operation of a filter sand installation, dark colored deposits appear at the surface of the filter mass. In the long term they have no consequence and disappear after a few filtration and wash cycles. If, after several weeks of filtration, these phenomena have not disappeared, it will be necessary to examine the filter sand. The elimination of fine sand must stop after 1 or 2 months of activity. If this sand continues to be carried away after the first several dozen washes it is necessary to reexamine the hydraulic criteria of the washing conditions, the granulometry of the filter mass, and the filter's resistance to shear and abrasion.

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Theory and Design Methodology

In this subsection, basic design theory for preliminary sizing and specifying equipment are reviewed. Some sample design calculations are included. References cited at the end of the chapter can be consulted for more detailed information and design methods. For solid-liquid separation methods, the reader should refer to ***Liquid Filtration***, 2nd edition, by N. P. Cheremisinoff, Butterworth-Heinemann Publishers (1998).

Gas Filtration: Vendors are the primary source for sizing baghouses, however the process engineer should at the very least make an estimate of the pressure drop requirements for an intended system. The pressure drop will, of course, provide information needed on the approximate size of the fan needed, and the energy requirements needed to operate a system. In fact, pressure drop is the primary parameter in establishing particle removal efficiency for a fabric filtration unit, and hence, is usually the starting place for discussions with the equipment supplier.

There are conflicting procedures for predicting pressure drop across fibrous filters. The three chief methods of predicting pressure drop are based on the hydraulic

radius theory, drag theory, and dimensional analysis. Experimental studies have shown dimensional analysis to be the most reliable, and it yields an equation for accurate pressure drop prediction. It is based on dimensional analysis of Darcy's law of flow through porous media and relates the pressure drop to the filter porosity. Darcy's original equation states:

$$\Delta PA / hq' = kw$$

where: A = cross-sectional area of filter

ΔP = pressure drop

h = filter thickness

q' = volumetric flowrate

k = permeability of the medium

w = fluid (gas) velocity

Darcy's empirical equations assume the gas to behave ideally; however, a modification of this is the following expression:

$$\Delta PA/hq' = k_3 w s_0^2 (1 - \epsilon)^2/\epsilon^3$$

where: k_3 = constant

s_0 = surface area/unit volume of solid material

ϵ = cloth porosity

From dimensional analysis, the following empirical expression is obtained:

$$k = \Delta PAd_e^2/hq'w = 64 (1 - \epsilon)^{1.5} (1 + 56 (1 - \epsilon)^3)$$

where: k = permeability coefficient

d_e = effective fiber diameter

This formula has proven accurate for fiber diameters ranging from 1.6 to 80 μ, and for filter porosities ranging from 0.700 to 0.994.

Another empirical expression that can be used to predict pressure drop, again based upon application of dimensional analysis is given by the following formula. This relationship is applicable to fabrics having porosities over the range of 0.88 to 0.96,

and for fiber diameters ranging from 0.1 to 0.3 μ. In this formula the parameters are defined as follows: λ is the mean-square fiber diameter and k" is the resistance coefficient.

$$\Delta PA\lambda/hq'w = k"(1 - \epsilon)^{1.5}$$

A final empirical expression that can be used is given below. This equation relates pressure drop to fabric properties which are readily obtainable from equipment suppliers.

$$\Delta P = w\xi(k_0 + k_1 W_A)$$

where: ξ = air to cloth ratio

k_0 = weave resistance coefficient

k_1 = cake resistance coefficient

W_A = weight of filtrate cake per unit area of surface

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Solid-Liquid Filtration: As already described, solid-liquid filtration operations handle sludges of varying characteristics ranging from granular, incompressible, free-filtering materials to slimes and colloidal suspensions that are compressible and tend to foul the filtering medium. The important characteristics of a sludge are its structure (granular and open or colloidal and dense) and its degree of compressibility. With an incompressible sludge, the resistance of the cake is essentially independent of the pressure. If the sludge is compressible, resistance increases rapidly with increasing pressure. In the process of suspension separation, a continuous accumulation of solids takes place, forming a wet filter *cake* and a *filtrate*. This type of filter operation provides a particle retention that exceeds that obtained based on the hole size of the filtering medium. Smaller particles tend to pass through the medium's pores. In contrast to this type of *filtering medium,* filtration in the depth of the pores of a filtering medium is responsible for capturing particles much smaller than the pore sizes in the actual passages. *Depth filtration* exhibits increased retentivity as particles travel from the upstream face of the filter plate to the downstream side. Fine particles are removed in the deeper portions of the filter, filling the pores. The most common form of filtering in the chemical and process industries is *cake filtration* on the filtering medium. This involves handling the permeability of a bed of porous material. Filtration is based on a complicated interaction between the sludge and the filtering medium. The solid particles entrained in the liquid flow through the filtering medium encounter different conditions. Particles larger than the pores on the filter media surface are captured.

Many particles whose sizes are less than that of a pore at its narrowest cross section tend to pass through the filter altogether, along will, the filtrate. However, a portion of these particles is captured inside the filter due to adsorption onto the walls of the pores, or by impingement in the course of its twisted flow path. Such an arrested particle decreases the effective flow cross section, while increasing the probability of retaining the particles that follow. Some particles simply become lodged in the pore, making it impenetrable for other particles. With high solids concentration sludges, even particles that are relatively small in comparison to the pore size may not enter into the media and will tend to remain on the filter surface forming "bridges" over individual openings in the filter plate.

Filtrate flows through the filter medium and cake because of an applied pressure, the magnitude of which is proportional to the filtration resistance. This resistance results from the frictional drag of the liquid as it passes through the filter and cake. Hydrostatic pressure varies from a maximum at the point where liquid enters the cake, to zero, where liquid is expelled from the medium; consequently, at any point in the cake the two are complementary. In other words, the sum of hydrostatic and compressive pressures on the solids always equals the total hydrostatic pressure at the face of the cake. As such, the compressive pressure acting on the solids vary from zero at the face of the cake to a maximum at the filter medium. When the space above the sludge is connected to a source of compressed gas, or the space under the filter plate is connected to a vacuum source, filtering is accomplished under a constant pressure differential, because the pressure in the receivers is maintained constant. In this case, the rate of the process decreases due to an increase in the cake thickness and, consequently, its resistance. If the sludge is fed to the filter with a reciprocating pump having constant capacity then filtering takes place under constant flowrate. In this case, the pressure differential increases due to an increase in the cake resistance. If he sludge is fed by a centrifugal pump, its capacity decreases with an increase in cake resistance and the filtering is under variable pressure differences and flowrates.

The most favorable filtration process with cake formation is the process in which there is no clogging of the filtering medium. Such a process is observed at sufficiently high concentrations of solid particles in sludge. This concentration may be assumed conditionally to be in excess of 1% by volume. To prevent pore clogging in the filtering medium when handling relatively low solids concentrations (e.g., 0.1 to 1% by volume), general practice is to increase the solids concentration in thickeners before the sludge is fed to the filter. To improve the filtration characteristics of sludges that are difficult to filter, *filter aids and/or flocculants* often are used to increase particle size. A filter aid is a finely divided solid material, consisting of hard, strong particles that are, in mass, incompressible. The most common filter aids (as an admix to the sludge) are diatomaceous earth, expanded perlite, solkafloc, fly ash or carbon. Filter aids may be applied in one of two ways. The first method involves the use of a precoat filter aid, which can be applied as a

thin layer over the filter before the sludge is pumped to the apparatus. This prevents fine sludge particles from becoming so entangled in the filter plate that the resistance of the filtering medium itself becomes high. Furthermore, it facilitates the removal of the cake at the end of the filtration cycle. The second application method involves the incorporation of a certain amount of the material with the sludge before introducing it to the filter. The addition of filter aids increases the porosity of the sludge, decreases its compressibility and reduces the resistance of the cake. An acceptable filter aid should have a much lower filtration resistance than the material with which it is being mixed.

It should also reduce the filtration resistance by two-thirds to three-fourths with the addition of no more than 25% by weight of filter aid as a fraction of total solids. The addition of only a small amount of filter aid (e.g., 5% of the sludge solids) can cause a significant increase in filtration resistance. When the amount of filter aid is so small that the particles do not interact, they form a coherent structure and resistance may be affected adversely. An increase in the particle size tends to improve the filtration process, which may be achieved by *flocculation*. Flocculants are filtered out more readily into a cake, which is permeable and does not clog the filtering medium. Flocculation is an application of the principles of colloid chemistry. Only when conditions are proper will particle flocs form. Mild agitation assists in bringing particles into contact for the action to take place. Many dispersions are often unstable and chemically ready to flocculate. Others require that the dispersing agent on their surfaces be neutralized or precipitated. There are others in which the particulate matter need to be flocculated with a precipitate (e.g., in lime-alum clarification of wastewater). Once chemical conditions are satisfied, flocculation action proceeds. The operation of chemical pretreatment consists of flocculation or coagulation, which involves the manipulation of charges on the particulates. Small particulates, which are highly charged, form stable suspensions that neither readily settle nor filter.

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The addition of trivalent cationic salts (e.g., $AlCl_3$, $FeCl_3$) or a change in the solution pH are common methods employed for changing the surface potential. Most natural substances are negatively charged. The addition of coagulants, in the form of salts, acids or bases, tends to neutralize surface charges, thus permitting attractive van der Waals-London-type forces to promote particle agglomeration. High-molecular weight linear polymers (which may be cationic, anionic or nonionic) combine with charged particulates to form large sedimenting flocs. Bentonite and kaolin clays are used principally as extender-aids in combination with flocculants. Because of active sites present on the extenders, they attach to nucleating sites. Their overall effect is to increase the density of the flocs and settling rates.

Filtration frequently is accompanied by hindered or free gravitational sedimentation of solid particles. The directions of action of gravity force and filtrate motion may be cocurrent, countercurrent or cross current, depending on the orientation of the

filtering plate, as well as the sludge location over or below the filtering plate. Particle sedimentation complicates the filtration process and influences the controlling mechanisms. Furthermore, these influences vary depending on the relative directions of gravity force and filtrate motion.

If the sludge is above the filter plate, particle settling leads to more rapid cake formation with a clear filtrate, which can be evacuated from the filter by decanting. If the sludge is under the filter plate, particle settling will prevent cake formation, and it is necessary to mix the sludge to maintain its homogeneity. When the cake structure is composed of particles that are readily deformed or become rearranged under pressure, the structures generally are termed compressible cakes. Those not readily deformed are called semicompressible, and those that deform only slightly are considered noncompressible. Porosity (the ratio of pore volume to volume of cake) does not decrease with increasing pressure differential. The porosity of a compressible cake decreases under pressure, and its hydraulic resistance to the flow of the liquid phase increases with pressure differential across the filter media. Cakes containing particles of inorganic substances, with sizes in excess of 100 μ, may be considered practically incompressible. Examples are sand particles and crystals of carbonates of calcium and sodium. The cakes containing particles of metal hydroxides, such as ferric hydroxide, cupric hydroxide and aluminum hydroxide, as well as sediments consisting of easily degradable aggregates, which are formed from primary fine crystals, are compressible. After completion of cake formation, treatment of the cake is dependent on the specific filtration objectives. For example, the cake itself may have no value, whereas the filtrate may.

Depending upon the disposal method and properties of the particles, the cake may be discarded in a dry form or as a slurry. In both cases, the cake is subjected to washing, following its formation, either immediately, or after a period of drying if the volume of wash liquor is limited, or to avoid filtrate dilution or excessive wash concentrations. In some cases a second washing is required, followed by a drying period in which filtrate must be removed from the cake; or wet discharge follows for disposal; or repulping and a second filtration occur; or dry cake disposal is preferable. Similar treatment options are employed in cases where the cake is valuable and all contaminating liquors must be removed, or in which both cake and filtrate are valuable. In the latter, cake-forming filtration is employed without washing, to dewater cakes in which a contaminating liquor forms the moisture content of the cake. Two basic methods of cake washing are displacement and dilution. Displacement involves pouring the washing liquid onto the cake surface in the form of a liquid layer or by means of atomization. The liquid passes through the pores , thereby displacing bound liquid in its path. The method is used with easily washed cakes that have no blind pores. In the dilution method, the cake is removed from the filter and mixed with the liquid. The suspension is then separated on the filter. This method is preferred with cakes that are difficult to wash.Design formulations developed for a filtration process analysis are based upon a two part

concept or approach. The first half consider flow within the cake, while the second examines the flows imposed on the cake and pumping system. The characteristics of the pump relate the pressure and the flowrate at the exit face of the cake. It is the cake resistance that determines the pressure drop. During filtration, liquid flows through the porous filter cake in the direction of a decreasing hydraulic pressure gradient. The porosity, ϵ, is at a minimum at the point of contact between the cake and filter plate, i.e., where $x = 0$, and it is a maximum at the cake surface ($x = L$) where the sludge enters. The drag imposed on each particle is transmitted to adjacent particles. Therefore, the net solid compressive pressure increases as the filter plate is approached, resulting in a decrease in porosity. It may be assumed that particles are in contact at one point only on their surface, and that liquid surrounds each particle completely. Hence, the liquid pressure acts uniformly in a direction along a plane perpendicular to the direction of flow. As the liquid flows past each particle, the integral of the normal component of force leads to the form drag, and the integration of the tangential components results in the frictional drag. If the particles are nonspherical, we still may assume single point contacts between adjacent particles.

We consider flow through a cake with the membrane located at a distance, x, from the filter plate. Neglecting all forces in the cake other than those created by drag and hydraulic pressure, a balance from x to L gives:

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$$F_s + AP_L = AP$$

where the applied pressure, P, will be a function of time but not of the distance, x. The term, F_s, is the cumulative drag on particles, increasing in the direction from $x = L$ to $x = 0$. As single-point contact is assumed, the hydraulic pressure, P_L, is effective over the entire cross section (A) of the cake. Dividing the above expression by A and denoting the compressive drag pressure by $P_s = F/A$, we obtain:

$$P_L + P_s = P$$

The term P_s is a fictitious pressure because the cross-sectional area, A, is not equal to either the surface area of the particles or the actual contact areas. In actual cakes, there is a small area of contact, A_c, and the solid pressure may be defined as F_s/A_c. Taking differentials with respect to x in the interior of the cake, we obtain:

$$dP_s + dP_L = 0$$

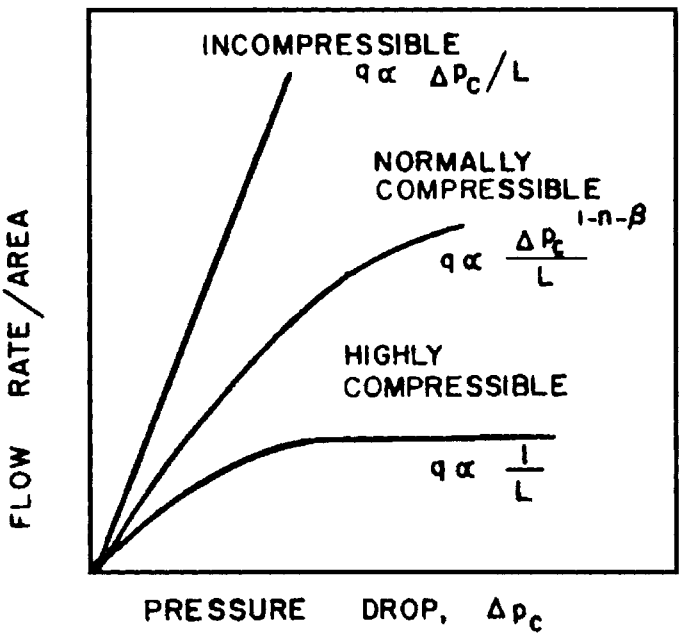

Figure. 30. *Flowrate - pressure drop relationships.*

This means that drag pressure increases and hydraulic pressure decreases as fluid moves from the cake's outer surface toward the filter plate. From Darcy's law, the hydraulic pressure gradient is linear through the cake if the porosity, ϵ, and specific resistance, α, are constant. The cake then may be considered "incompressible." This is shown by the straight line on the plot of flowrate per unit filter area versus pressure drop in Figure 30. The variations of porosity and specific resistance are accompanied by varying degrees of compressibility, also shown in Figure 30.

Because pore sizes in the cake and filter plate are small and the liquid velocity through the pores is low, the flow of filtrate may be considered laminar and Poiseuille's law is applicable. Filtration rate is directly proportional to the difference in pressure and inversely proportional to the fluid viscosity, as well as to the hydraulic resistance of the cake and filter plate. Because the pressure and hydraulic resistances of the cake and filter plate change with time, the variable rate of filtration may be expressed as:

$$u = dV/Ad\tau$$

where V = volume of filtrate, A = filtration area (m^2), and τ = filtration time (s).

Assuming laminar flow through the filter channels, the basic differential equation of filtration is simply stated as follows:

$$u = dV/Ad\tau = \Delta P/[\mu(R_c + R_f)]$$

where ΔP = pressure difference (N/m^2), μ = viscosity of filtrate ($N\text{-}s/m^2$), R_c = filter cake resistance (liter/min), R_f = initial filter resistance (resistance of filter plate and filter channels, liter/min), u = filtration rate (m/s) (filtrate flow through cake and filter plate), and $dV/d\tau$ = filtration rate (m^3/s) filtrate flow. The value of R_f may be assumed constant during filtering. This assumption is based on any increase due to solids penetrating inside the pores. Filter cake resistance, R_c, is the resistance to filtrate flow per square meter of filtration area. R_c increases with cake thickness during filtration. At any instant, R_c depends on the mass of solids deposited on the filter plate as a result of the passage of V m^3 of filtrate. To determine the relationship and residence time, τ, the above expression must be integrated. This enables R_c to be expressed in terms of V.

Defining the ratio of cake volume to filtrate volume as x_o, then the cake volume is x_oV. The cake volume also may be expressed by the product h_cA, where h_c is the cake height in m. Hence,

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$$x_oV = h_cA$$

Hence, the thickness of the cake, uniformly distributed over the filter plate, is

$$hc = x_o(V/A)$$

And the filter cake resistance may be expressed as

$$R_c = r_ox_o\,(V/A)$$

where r_o is the specific volumetric cake resistance, m^{-2}. As follows from this expression, r_o characterizes the resistance to the liquid by a cake of 1-m thickness. Substituting for R_c into this expression, we obtain:

$$dV/Ad\tau = u = \Delta P/[\mu\{r_ox_o(V/A) + R_f]$$

Parameter x_o can be expressed in terms of the ratio of the mass of solid particles settled on the filter plate to the filtrate volume, x_w, and, instead of r_o, a specific mass cake resistance, r_w, is used. That is, r_w is the resistance to the flow presented by a uniformly distributed cake in the amount of 1 kg/m^2. Replacing units of volume by mass, the term $r_o x_o$ into the above expression changes to $r_w x_w$. Neglecting the filter plate resistance (i.e., $R_f = 0$), then:

$$r_o = \Delta P/\mu h_c u$$

At $\mu = 1$ N-s/m^2, $h_c = 1$ m and u = 1 m/s, the value $r_o = \Delta P$. Thus, the specific cake resistance equals the pressure difference required by the liquid phase (with a viscosity of 1 N-s/m^2) to be filtered at a rate u = 1 m/s for a cake 1 m thick. This hypothetical pressure difference is, however, beyond a practical range. For highly compressible cakes, the value ro reaches 10^{12} m^{-2} or more. Assuming V = 0 (at the start of filtration) where there is no cake over the filter plate, the equation becomes:

$$R_f = \Delta P/\mu u$$

At $\mu = 1$ N-s/m^2 and u = 1 m/s, $R_f = \Delta P$. This means that the filter plate resistance is equal to the pressure difference necessary for the liquid phase to pass through the filter plate at a rate of 1 m/s. For many filter plates, R_f is typically 10^{10} m^{-1}.

With the above as an introduction, we now consider the important operational case of filtration performed under constant pressure. In practice, all the parameters defined above are nearly constant under steady state conditions except V and τ, which are varied by the operator. We may therefore integrate the working expression for filtration over the limits of volume from 0 to V, and for residence time over the limits of 0 to τ:

$$\int \mu(r_o x_o(V/A) + R_f)dV = \int \Delta PA d\tau$$

Upon integration and reworking the relationship (details may be obtained from ***Fluid Mechanics and Unit Operations***, D. S. Azbel and N. P. Cheremisinoff, Ann Arbor Science Publishers, Mich. (1983)), we obtain the following working formula:

$$V^2 + 2(R_f A/r_o x_o)V = (2\Delta P A^2/\mu r_o x_o)\tau$$

This expression shows the relationship between filtration time and filtrate volume. The equation is applicable to both incompressible or compressible cakes, because at constant ΔP the values r_o and x_o are constant. For constant ΔP, an increase in the filtrate volume results in a reduction in the filtration rate. If we assume a definite filtering apparatus and set up a constant temperature and filtration pressure, then the values of R_f, r_o, μ and ΔP will be constant. We now take note of the well-known filtration constants K and C, which are derived from the above expressions:

$$K = 2\Delta PA^2 / \mu r_o x_o$$

$$C = R_f A / r_o x_o$$

The filtration process may therefore be expressed by a very simple relationship:

$$V^2 + 2VC = K\tau$$

Constants K and C can be readily obtained from experiments conducted on a prototype machine, from whence the volume of filtrate obtained for a definite time interval (for a specified filter, at the same pressure and temperature) can be calculated. If process parameters are varied, new constants K and C can be estimated from the above expressions. The last expression can be further modified by denoting the constant τ_o as $= C^2/K$, and substituting:

$$(V + C)^2 = K(\tau + \tau_o)$$

This equation is then the expression describing the process of filtration under constant pressure. The form of the relationship is parabolic. From a plot of $(V + C)$ versus $(\tau + \tau_o)$, the filtration process may be represented by a parabola with its apex at the origin, as shown in Figure 31. Moving along the axes to point C and τ_o provides the characteristic filtration curve for the system in terms of volume versus time. Because the parabola's apex is not located at the origin of this new system, it becomes clear why the filtration rate at the beginning of the process will have a finite value, as seen in actual practice. It is often overlooked that constants C and τ_o have physical significance. They are equivalent to a fictitious layer of cake of equivalent resistance. The formation of this fictitious cake follows the same parabolic relationship. Parameter τ_o denotes the time required for the formation of this fictitious cake, and C is the volume of filtrate required. Upon integrating the filtration expression and rearranging terms, a more convenient linear form of the expression is derived, lending itself to a graphical analysis.

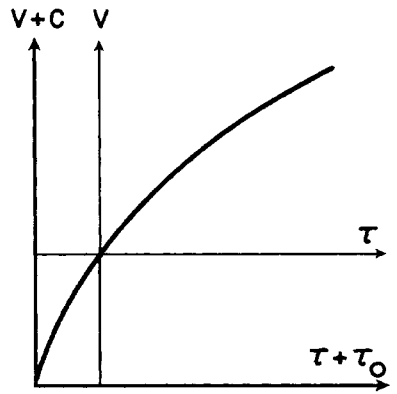

Figure 31. *The typical filtration curve.*

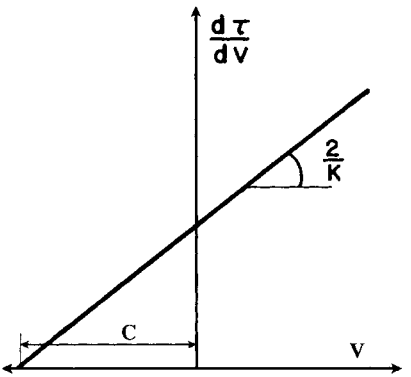

Figure 32. *Linear form of equation.*

The linear analytical expression is:

$$d\tau/dV = 2V/K + 2C/K$$

A plot of this expression is provided in Figure 32, where the slope of the straight line is 2/K, and the intercept value is C. Experimental determination of $d\tau/dV$ is straightforward. Filtrate volumes V_1 and V_2 are measured at time intervals τ_1 and $\tau 2$, respectively. From the linear equation of filtration, the quotient $(\tau_2 - \tau_1)/(V_2 - V_1)$ is:

$$(\tau_2 - \tau_1)/(V_2 - V_1) = (2/K)[(V_1 + V_2)/2] + 2C/K$$

Constants C and K can be determined from several measurements of filtrate volumes taken at different time intervals. There are some doubts as to the actual constancy of C and K during constant pressure filtration. Constants C and K depend on r_o (specific volumetric cake resistance), which, in turn, depends on the pressure drop across the cake. This ΔP causes some changes in the cake, especially during the initial stages of filtration. When the cake is very thin, the main portion of the total pressure drop is exerted on the filter medium. As the cake becomes thicker, the pressure drop through the cake increases rapidly but then levels off to a constant value. Isobaric filtration shows insignificant deviation from the expressions developed. For approximate calculations, it is possible to neglect the resistance of the filter plate, provided the cake is not too thin. Then the filter plate resistance, R_f, is equal to zero, $C = 0$, and $r_o = 0$. Hence, a simplified equation is: $V^2 = K\tau$.

The following example helps to illustrate the use of the equations presented up to this point. *An aqueous slurry was filtered in a small laboratory filter press with a pressure drop of 0.5 atm and at a temperature of 20 °C. After 10 minutes, 4.7 liters of filtrate were obtained; after 20 minutes, 7.0 liters were collected. From experiments at other pressures, it was determined that the cake compression coefficient was s = 0.4. We wish to determine the volume of filtrate expected after 30 minutes from a filter press having a filtering area 10 times greater than the laboratory press if the filtration is to be performed at 1.5 atm pressure. The liquid temperature will be 55 °C. We also wish to determine the rate of filtration at the end of the process.*

The solution to this problem is as follows. Evaluate K and C from the experimental data at $\Delta P = 0.5$ atm. through the use of the expression $V^2 + 2VC = K\tau$. For V = 4.7 liters and $\tau = 10$ min, we obtain:$(4.7)^2 + 2 \times 4.7 \times C = 10K$. For V = 7.0 liters and $\tau = 20$ min, we obtain: $(7.0)^2 + 2 \times 7 \times C = 20K$. Solving, we obtain C = 1.04 and K = 3.18. These coefficients change as a function of the filtration area, temperature and pressure. We note that the following empirical expression holds:

$$r_o = a\Delta P^s$$

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where a is a coefficient. Constant C may be written in the following form:

$$C = (R_f/ax_o) \times (A/^{\Delta ps})$$

The term r/ax_o is a constant and independent of filtration conditions. We find that the ratio of constant C_2 at pressure 1.5 atm. to constant C_1 for the process under pressure at 0. 5atm. is as follows: $C_2/C_1 = (A_2/A_1) \times (\Delta P_1/\Delta P_2)^s$. Then $A_2/A_1 = 10$ and $s = 0.4$, and hence, we solve for C_2: $C_2 = 1.04 \times 10(0.5/1.5)^{0.4} = 8.85$. Coefficient K is determined from:

$$K = (2/ax_o) \times \Delta P^{1-s} A^2/\mu;$$

$$K_2/K_1 = (\Delta P_2/\Delta P_1)^{1-s} (A_2/A_1)^2 (\mu_1/\mu_2)$$

The viscosity of water at 20 °C is $\mu_1 = 1$ cp; at 55 °C, it is 0.5 cp. Therefore, $_{K2} = 3.18 \times (1 .5/0.5)^{1-0.4} \times 100 \times (1/0.5) = 1230$. From the values of K_2 and C_2, we determine $\tau_o = (8.85)^2/1230 = 0.0637$. Applying the linear expression of filtration then: $(V + 8.85)^2 = 1230(\tau + 0.0637)$. For time $\tau = 30$ min, we obtain

volume V in liters: $V = \{1230(30 + 0.064)\}^{1/2} - 8.85 = 183$ liters. The final filtration rate (after 30 minutes) is obtained from $dV/d\tau = K/[2(V + C)] = 1230/[2(183 + 8.85)] = 3.2$ liters/min. The average rate of filtration is: $V/\tau = 183/30 = 6.1$ liters/min.

We now turn attention towards the case of constant-rate filtration. When sludge is fed to a filter by means of a positive displacement pump, the rate of filtration is nearly constant, i.e., $dV/d\tau$ = constant. During constant-rate filtration, pressure increases with cake thickness. As such, the principal filtration variables are pressure ind filtrate volume, or pressure and filtration time. Integrating the filtration equation for a constant-rate process, we find that the derivative $dV/d\tau$ can simply be replaced by V/τ, and we obtain:

$$\Delta P = \mu r_o x_o (V^2/A^2\tau) + \mu R_f (V/A\tau)$$

For incompressible cakes, coefficient r_o is constant and independent of pressure. For compressible cakes ($s \neq 0$) r_o may be estimated from the expression $r_o = a\Delta P^s$. Substituting for r_o into the above relation, we obtain:

$$\Delta P = \mu a x_o \Delta P^s (V/A\tau)^2 \tau + \mu R_f (V/A\tau)$$

Filtration experiments in a prototype machine at constant pressure or constant rate permit determination of ax_o, as well as s and R_f, for a given sludge and filtering medium. Consequently, it is possible to predict the time required for the pressure drop to reach the desired level for a specified set of operating conditions. In the initial stages of filtration, the filter medium has no cake. Furthermore, ΔP is not zero but has a certain value corresponding to the filter medium resistance for a given rate. This initial condition is:

$$\Delta P_o = R_f \mu (V/A\tau)$$

For an incompressible cake ($s = 0$), the equation takes the form:

$$\mu a x_o (V/A\tau)^2 + R_f \mu (V/A\tau) = \Delta P$$

As follows for the filtration of incompressible sediment (at a constant rate), the pressure increases in a direct proportion to time. However, the above equation

shows that pressure increases faster than time. In some cases the resistance of the filter medium may be neglected, especially for thick cakes. Assuming $R_f = 0$, our equation simplifies to the following:

$$\Delta P^{1-s} = \mu x_o a(V/A\tau)^2\tau$$

An increase in pressure influences not only coefficient r_o, but also the cake's porosity. Cake on the filter plate is compressed; consequently, the additional liquid is squeezed out of the cake. Thus, along with a constant feed of sludge, the flowrate will not be stable and will change somewhat in time. The weight of dry solids in a cake is $W' = x_o'V$, where x_o' is the weight of solids in the cake per unit of filtrate volume. The concentration of solids in the feed sludge is expressed by weight fraction, c. It also is possible to determine experimentally the weight ratio of wet cake to its dry content, m. Hence, a unit weight of sludge contains $m \times c$ of wet cake. Denoting γ as the specific weight of filtrate, then $(1 - mc)/\gamma$ is the filtrate volume per unit weight of feed sludge. But this unit contains c amount of solids, and the ratio of solids in the cake to the filtrate volume is defined as:

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$$x_o' = c\gamma/(1 - mc)$$

Thus, from sludge concentration c and the weight of a wet cake per kg of dry cake solids, x_o' can be determined. If the suspension is dilute, then c is small and the product of $m \times c$ is small. This means that x_o' will be almost equal to c. Since x_o' depends on the product mc, at relatively moderate suspension concentrations this effect will not be great and thus may be neglected. However, for the filtration of concentrated sludges, the mentioned phenomena will play some role, i.e., at constant feed the filtrate changes with time.

Variable-rate, variable-pressure filtration is also an important operating mode for filters. Figure 33 illustrates typical pressure profiles across a filter. This figure shows the distribution of static pressure, P_{st}, in liquid and P along the cake thickness and filter plate: (I, II) are boundaries between the cake and sludge at τ" and τ'; (III, IV) are boundaries between the cake layers or cake and filter plate at τ' and τ"; (V) is the boundary line between the cake and filter plate or the free surface of the filter plate. Curves (1, 3) represent $P_{st} = f(h_{oc})$ at τ', and curves (2, 4) are $P_{st} = f(h_{oc})$ and $P = f(h_{oc})$ at τ". The compression force in the cake section is defined as: $P = P_1 - P_{st}$. In this statement P_1 is the pressure acting on the sludge over the entire cake thickness and P_{st} is the static pressure over the same section of cake. Pressure P corresponds to the local specific cake resistance $(r_w)x$. At the sludge-cake interface, $P_{st} = P_1$ and $P = 0$; and for the interface between the cake and filter

plate, $P_{st} = P_{st}'$ and $P = P_1 - P_{st}$. P_{st}' corresponds to the resistance of filter plate, ΔP_f, and is expressed by the equation: $\Delta P_f = \mu R_f W$; where W is defined as the filtration rate (m^3/m^2-s). Not that ΔP_f is constant during the filtration process, and that P is also the driving force of the process. We consider a small element of solid particles weight in a cake of unit area as $x_w dq$, where q is the filtrate volume from 1 m^2 of filtering area. The incremental pressure drop for this element is:

$$dP/x_w dq = \mu(r_w)_x W$$

Parameter xw undergoes relatively small changes with P. In practice, and average value can be assumed. Note that W is constant for any section of the cake. The above expression may be integrated over the cake thickness for the limits of P = 0 to $P = P_1 - P_{st}$, and from q = 0 to q = q.

$$q = (1/\mu x_w W)\int dP/(r_w)_x$$

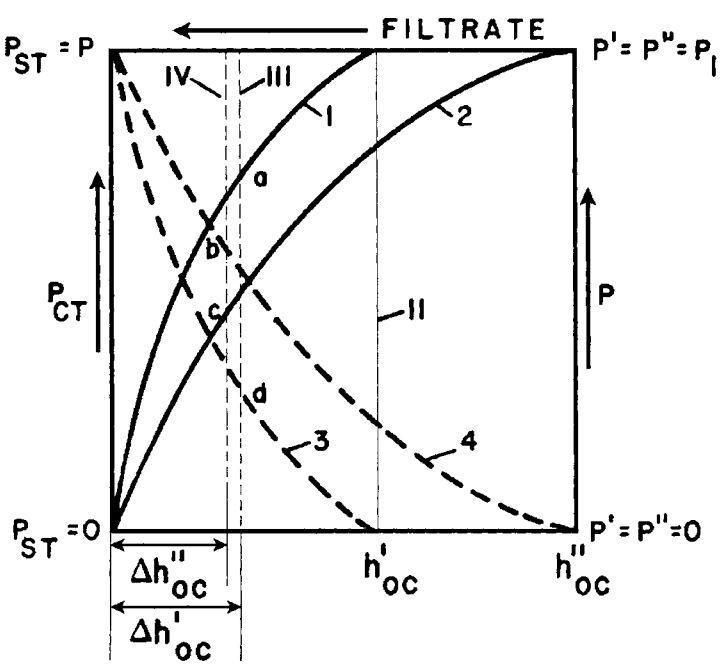

Figure 33. *Pressure profiles across the filter cake and filtrate .*

Parameters q and W are variables when filtration conditions are changed. Coefficient $(r_w)_x$ is a function of pressure; $(r_w)_x = f(P)$. The exact relationship can be derived from experiments in a device called a compression-permeability cell. Once this relationship is defined, the integral of the right hand side of the above equation may be evaluated analytically. Or, if the relationship is in the form of a curve, the evaluation may be made graphically. The interrelation between W and P, is established by the pump characteristics, which define $q = f(W)$ in the integral. Filtration time may then be determined from $dq/d\tau = W$, from which we may state:

$$\tau = \int dq/W$$

The following example illustrates the principles behind variable rate, variable-pressure filtration. *The following data were obtained from separation tests for a particular suspension using a filter: $\mu = 0.00148$ N-s/m^2, $x^w = 3.05$ kg/m^3 and $R_f = 6.56 \times 10^{10}$ m^{-1}. The relationship $(1/r_w)_x = f(P)$ was obtained from a compression-permeability cell and the results are given by curve 1 in Figure 34. The pump characteristics in terms of capacity per unit filter area are given by curve 1 in Figure 35. Determine the relationship between filtrate volume and the time of filtration.* The solution to this problem is as follows. According to the expression $\Delta P_f = \mu R_f W$, the static pressure at the interface between the cake and filter plate is numerically equal to the plate resistance provided that the pressure exerted on the free surface of the filter plate is equal to zero. Hence: $P_{st}' = \Delta P_f = 0.00148 \times 6.56 \times 10^{10} W = 97.1 \times 10^{10} W$.

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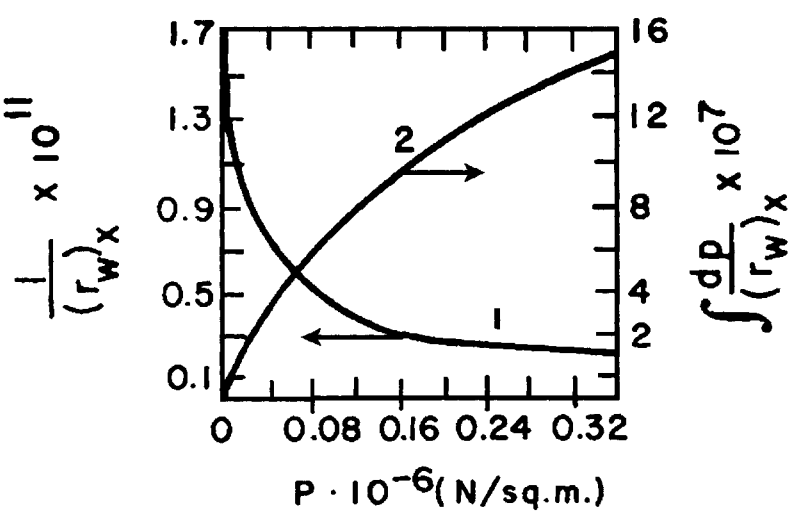

Figure 34. *Specific resistance $1/(r_w)_x$ vs. P (curve 1) and integral (curve 2).*

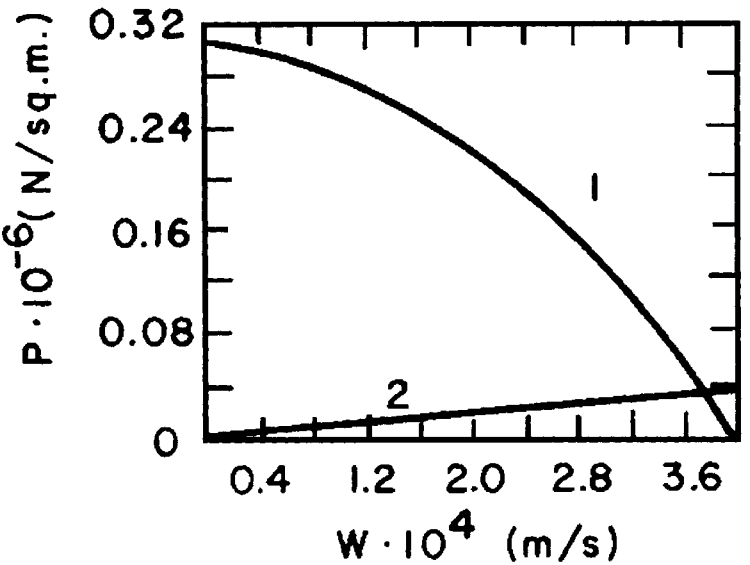

Figure 35. *Characteristic pump curve (1) and liquid static pressure on the boundary between the cake and filter plate (curve 2).*

Curve 2 in Figure 35 shows the relationship of liquid static pressure on the boundary between the cake and filter plate versus filtration rate. As seen from Figure 35, at any filtration rate, P_1 is spent for overcoming the resistance of filter plate P_{st}' and cake resistance. The intersection of curve 1 and line 2 corresponds to the start of filtration when there is no cake formed and the total pressure developed by the pump is spent for overcoming the resistance of the filter plate. It is assumed in this case that the filter plate resistance is independent of pressure. If it depends on pressure, the design should consider some average value, $P_{st}' = f(W)$, which may be determined from an experiment. For the limits from $P = 0$ to the increasing values $P = P_1 - P_{st}'$ (refer to Figure 34), curve 2 is obtained by means of graphical integration to show the dependence of the integral from forces compressing the cake. To determine q versus W at given y and x_w, we examine Figure 35 to find the values W_1P_1 and $P_1 - P_{st}'$; then for each value of $P - P_{st}' = P$ we determine from curve 2 the value of the integral. From this information, q can be determined. The results of these calculations are given in Figure 36. To determine $\tau = f(q)$, use Figure 36. This determines the value of the integral expression and, consequently, the value of τ. The data obtained in this manner are plotted in Figure 37. A more in-depth analysis of this case cane be found in ***Fluid Mechanics and Unit Operations***, D. S. Azbel and N. P. Cheremisinoff, Ann Arbor Science Publishers, Mich. (1983). Additional references on the subject are also given in this reference.

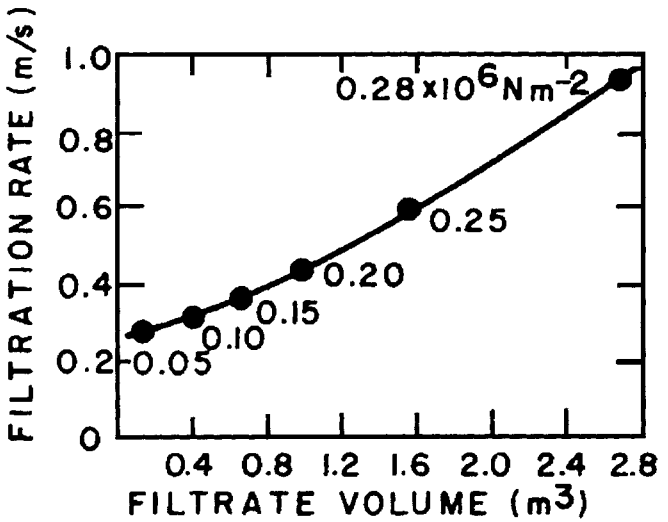

Figure 36. *Filtration rate vs. filtrate volume.*

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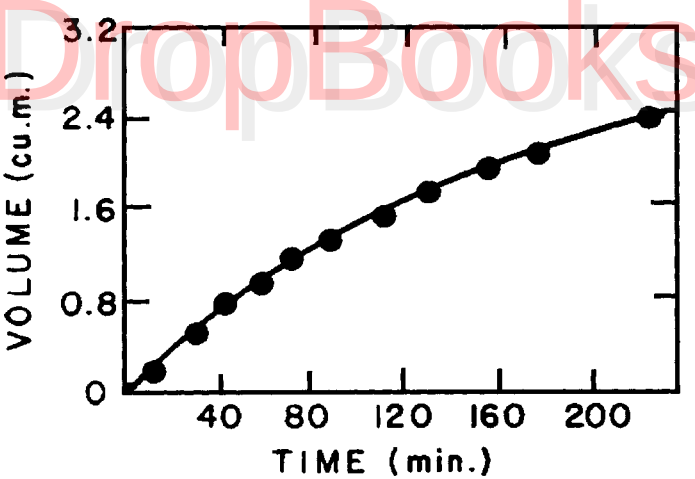

Figure 37. *Volume versus filtration time.*

The final method of operation to consider is that of filtration at constant pressure and constant rate. This method is employed when pure liquid is filtered through a cake of constant thickness and a constant pressure differential exists across the cake. Cake washing by displacement may be considered as filtering of a washing liquid through a constant cake thickness at constant pressure and flowrate. The rate of

washing is related to the rate of filtration during the final stages. Usually filtration during the final stages is performed under constant pressure. As such, expressions presented earlier are applicable. From filtration constants K and C, at constant pressure for a given system the filtration rate for the final period is determined. If the washing liquid passes through the filter following the same paths as the sludge and filtrate, then the difference between the washing rate and the filtration rate for this last period will mostly be due to a difference in viscosities of the wash liquor and the filtrate. Hence the expression $dV/d\tau = K/2(V + C)$ is applicable using the viscosity of the washing liquid, μ_w Denoting the rate of filtration during the last period as $(dV/d\tau)_f$, then the washing rate cane be determined by:

$$(dV/d\tau)_w = (dV/d\tau)_f \, \mu/\mu_w$$

Designing for Optimum Filtration

The preceding discussions have focused on an analysis of the most common operating modes for solid-liquid filtration. The analysis however is general, and does not allow practical calculations without further development of working expressions and more careful consideration of proper boundary conditions and assumptions. We now focus on a series of special formulas developed specifically for assessing optimum conditions of filtration. These offer simplified calculation methods that can be readily performed on a PC, and in some cases with a hand-held calculator.

Optimization implies performing maximum filtration (removal efficiency and capacity) under the most economically favorable conditions (i.e, considering capital investment, O&M costs, labor costs, etc.). We will only provide formulas for filtration capacity. One of the tasks of the process engineer should be to develop sufficient vendor and operating data that will enable a careful cost analysis to be made for a filtration system. To increase filtration capacity, it is necessary to eliminate the cake from the filter plate as quickly as possible. However, this approach is only practical for continuous operations. In batch operations, the separation process must be interrupted to remove the cake. The operating cycle of a batch filter consists of a series of steps, namely, filter preparation, sludge loading, filtration, cake washing, drying and scraping. Filtration, washing and drying are basic operations whose duration time increases with filtrate volume and cake thickness. Filter preparation, sludge loading and removing are auxiliary operations, whose duration may be assumed to be independent of the filtrate volume and cake thickness. For any given filter, the duration of auxiliary operations is practically constant. Such an assumption can result in appreciable error in estimating the maximum filter capacity in some cases. To increase the capacity of batch filters

(with respect to the basic operations) it is sound practice to repeat the cycle as often as possible. This can be accomplished by feeding the filter with small portions of sludge. However, frequent filtration cycles mean an equal frequency of auxiliary operations, which is obviously time consuming and costly. Consequently, an optimum cycle duration that provides the maximum filter capacity must be determined for each application. In considering the general case for evaluating the maximum filter capacity at constant pressure, the operating cycle can be viewed as having four steps in series: filtration, washing, cake drying and removal. The filtration step has already been described by the formula:

$$V^2 + 2(R_f A/r_o x_o)V = (2\Delta P A^2/\mu\, r_o x_o)\tau$$

But the following empirical expression can be applied to describing washing:

$$G/G_o = 1 - 0.25/(V_{wf}/V_o)$$

where G = weight of dissolved substances in the washing liquid, G_o = weight of dissolved matter in filtrate retained in the cake before washing, V_{wf} = volume of washing liquid, and V_o = filtrate volume in the pores before washing.

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The following equation describes the drying stage:

$$\tau_{dew}/C_{dew} = \{[(1 - m_o)^2 + 1]/2\} \times (m_e^{1-y})/(y - 1)$$

where τ_{dew} = dewatering time (s), C_{dew} = dewatering factor (s), m_o = residual cake saturation with liquid equal to the ratio of stationary liquid in the end of dewatering to the pore volume (fractions of unity), m_e = effective saturation with liquid equal to ratio of moving liquid volume to the total volume the same liquid and air (fractions of unity), and y = exponent, depending on the particle size in the cake. C_{dew} is the dewatering factor, that is defined as follows:

$$C_{dew} = \epsilon\mu r_o h_c^2/\Delta P$$

The dewatering factor, C_{dew}, combines the variables influencing the motion of liquids in the cake pores. The exponent y has been evaluated experimentally and literature-reported values range from 2.0 (for particles 1.8 mm in size) to 3.0 (for particles 0.09 mm in size). An average value of 2.5 may be assumed. A plot of

τ_{dew}/C_{dew} versus m_s for different values of m_o is shown in Figure 38. In this plot, τ_{dew}/C_{dew} is determined from the above equation at y = 2.5 and corresponding values of m_s (m_s = cake saturation with liquid, which is equal to the ratio of total stationary and moving liquid to pore volume, and is expressed in units of fraction of unity). This parameter can be calculated from the following equation:

$$m_s = (m_e - 2m_e m_o + m_o)/(1 - m_e m_o)$$

The curves shown permit evaluation of cake saturation by liquid rate versus dewatering time for a given set of conditions. During the dewatering stage, the volume of air sharply increases from zero to a maximum corresponding to the situation of air blowing through a dry cake. The amount of air passing through the pores as a function of dewatering time is given in Figure 39. In evaluating the optimum filtration time, we shall assume that the filter plate resistance is negligible. Furthermore, the duration of each stage will be evaluated as a function of filtrate volume on the assumption that all filtration conditions are constant. The time of filtration may be computed from earlier expressions.

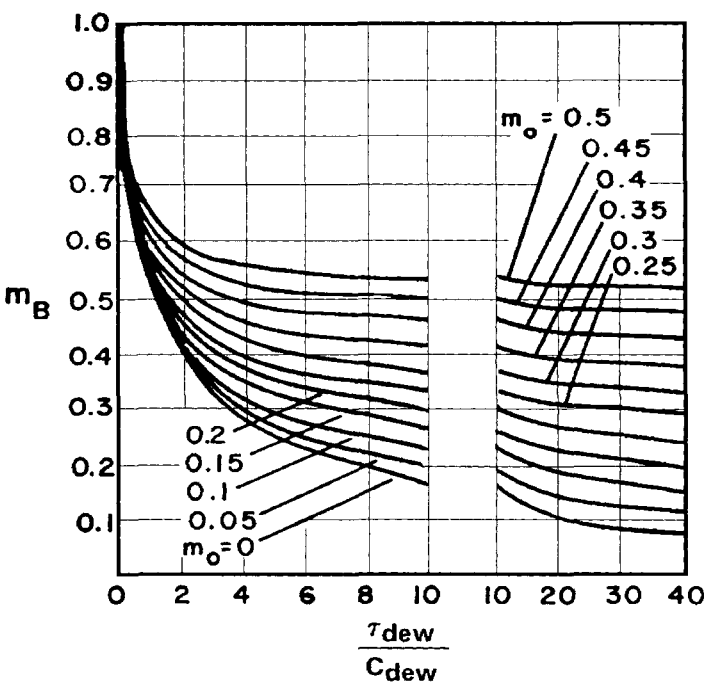

Figure 38. *Plot for determining cake saturation as a function of dewatering time.*

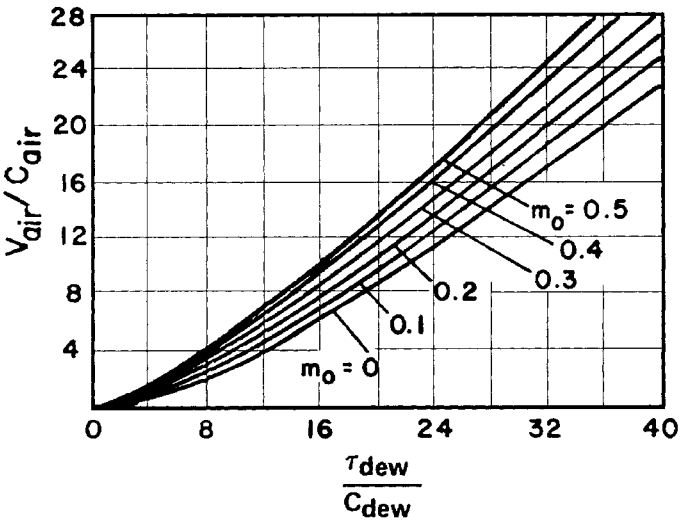

Figure 39. *Volume of blowing air vs. dewatering time.*

In the analysis we assume $R_f = 0$, and then $\tau = \frac{1}{2}(Bq^2)$, where $B = \mu r_o x_o/\Delta P$. The following relationships were developed and described by Azbel and Cheremisinoff (***Fluid Mechanics and Unit Operations***, Ann Arbor Science Publishers (1983)):

$$q_{wf} = 0.25q_o/(1 - G/G_o)$$

As $q_o = \epsilon x_o q$, then $q_{wf} = D"q$, where:

$$D" = 0.25\ \epsilon x_o/(1 - G/G_o)$$

Since the washing stage is essentially the case of filtration at constant cake thickness, q may be replaced by q_{wf}, and assuming the viscosity of the washing liquid to be the same as that of the filtrate, the following relationship between washing time and filtrate amount is developed:

$$\tau_w = BD"^2q^2$$

Drying time as a function of filtrate amount may be estimated from:

$$\tau_{dew} = BC'q^2$$

where parameter C' is defined as:

$$C' = \tfrac{1}{2} \times \{(1 - m_o)^2 + 1\} \times \{(m_e^{1-y} - 1)/(y - 1)\} \times (\epsilon x_o)$$

The total time required for all three steps is:

$$\tau_b = \tau + \tau_w + \tau_{dew} = Eq^2$$

This may also be stated as:

$$q = (\tau_b/E)^{1/2}$$

where $E = B(\tfrac{1}{2} + D''^2 + C')$. The average filter capacity per unit time for a total filtration cycle may be characterized by a conditional average filtration rate obtained by dividing filtrate volume per cycle per unit filter area by the total cycle duration ($\tau_t = \tau_b + \tau_{aux}$):

$$u_{cond} = (\tau_b/E)^{1/2}/(\tau_b + \tau_{aux})$$

Assuming the filter plate resistance is negligible, the following expression applies for filtration without washing and drying steps:

$$\tau_b = \tau_{aux} + 2(\mu R_f^2 \tau_{aux}/2\Delta P r_o x_o)^{1/2}$$

This expression can be represented graphically in dimensionless form to simplify its use. It is generally expressed as the so-called "filtration number", defined as follows: $F_{\ell 1} = \mu R_f^2 / 2\Delta P \tau_{aux} r_o x_o$. The filtration number, $F_{\ell 1}$, is dimensionless and varies from zero at $R_f = 0$ to a large value when there is an increase in the viscosity of the sludge and R_f or a decrease in pressure drop, auxiliary time, specific cake resistance and the ratio of cake volume to filtrate volume. It may be assumed in practice that $F_{\ell 1} = 0$ to 10. If washing and drying times are constant and independent of filtration time, they may be added directly to the auxiliary time. In

this case, the calculation of maximum filter capacity is similar to the case without washing and drying steps. From the times of the basic filtration steps, the filtrate volume per cycle can be computed along with the corresponding maximum capacity for different cases. The following equation can be used to determine the filtrate volume per cycle along with the corresponding maximum capacity for different trial cases in an evaluation:

$$q^2 + 2R_f q/r_o x_o = 2\Delta P\tau/\mu r_o x_o$$

For a cycle consisting of filtration and washing stages, and accounting for the filter plate resistance, the cake resistance corresponding to a maximum filter capacity is given by:

$$h_c = [2\Delta P x_o \tau/_{\mu ro}(1 + 2x_o L)]^{1/2}$$

where $L = \{\log(c_a/c_o)/K'_w\} \times \{\mu_{wf}/\mu\}$. In this parameter, c_a is the instantaneous concentration in washing liquid, and c_o is the concentration of solute in filtrate, K_w' is a constant which depends on the properties of the cake, the filtrate and the washing liquid, and hence must be determined from pilot tests. The following sample calculation summarizes the use of the major formulas.

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Design Example for a Batch Nutsche Filter Operation: A batch Nutsche filter operates under constant pressure and maximum capacity with three stages: filtration, washing and drying. The Nutsche filter with a 1 m^2 filtration area operates under the following set of conditions:

- Viscosity of filtrate and washing liquid μ (N-s/m^2) = 10^{-3}
- Air viscosity, μ_{air} (N-s/m^2) = 1.83 X 10^{-5}
- Surface tension of washing liquid, σ (N/m) = 0.04
- Cake specific resistance, r_o (m^{-2}) = 2 × 10^{12}
- Ratio of cake volume to filtrate volume, x_o = 0. 1
- Pressure drop, ΔP (N/m^2) = 40,000
- Porosity, ϵ = 0.45
- Weight ratio G/G_o dissolved substance extracted by washing liquid to the substance in the cake before washing = 0.98
- Residual liquid saturation of cake m_o (approximate, considering the specific cake resistance) = 0.5

- Effective cake saturation with liquid at the end of drying, $m_e = 0.1$
- Exponent y in dewatering equations = 2.5
- Auxiliary time of operations, τ_{aux} (min.) = 30
- Maximum permissible cake thickness, h_c (m) = 0.05

The resistance of the filter plate is negligible. The problem is to determine the filtration, washing and drying times, the cake thickness, the volumes of filtrate, washing liquid and drying air. The solution to this design case is outlined below in steps.

1. The constant E is first determined:

$B = \mu r_o x_o/\Delta P = [10^{-3} \times 2 \times 10^{12} \times 0.1]/40{,}000 = 5 \times 10^3$

$D'' = 0.25\, \epsilon x_o/(1 - G/G_o) = [0.25 \times 0.45 \times 0.1]/(1 - 0.98) = 0.56$

$C' = \frac{1}{2} \times \{(1 - m_o)^2 + 1\} \times \{(m_e^{1-y} - 1)/(y - 1)\} \times (\epsilon x_o) = 0.575$

$E = B(\frac{1}{2} \times D''^2 + C') = 5 \times 10^3(0.5 + (0.56)^2 + 0.575) = 6{,}945$

2. The filtrate volume is obtained, assuming $\tau_b = \tau_{aux}$, which corresponds to the maximum filter capacity: $q = (\tau_b/E)^{1/2} = (30 \times 60/6{,}945) = 0.51\ m^3$

3. Cake thickness: $h_c = x_o q = 0.10 \times 0.51 = 0.051$ m. This corresponds to the maximum cake thickness.

4. Next determine the filtration time:

$\tau = 1/2Bq^2 = 0.5 \times 5 \times 10^3(0.51)^2 = 650$ s

5. The volume of washing liquid: $q_{wf} = D''q = 0.56 \times 0.51 = 0.286\ m^3$

6. The washing time is: $\tau_w = B \times D''^2 \times q^2 = 5 \times 10^3 \times (0.56)^2 \times (0.51)^2 = 410$ s.

7. Determine the drying time: $\tau_{dew} = B \times C' \times q^2 = 5 \times 10^3 \times 0.575 \times (0.51)^2 = 745$ s. The total time of the filtration, washing and dewatering steps is approximately equal to the auxiliary time.

8. The amount of drying air is:

$C_{dew} = \epsilon \mu r_o h_c^2/\Delta P = [0.45 \times 10^{-3} \times 2 \times 10^{12} \times (0.051)^2]/40{,}000 = 58.5$ s. The ratio $\tau_{dew}/C_{dew} = 745/58.5 = 12.7$. From Figure 39, at mo = 0.5 and for $\tau_{dew}/C_{dew} = 12.7$, $V_a/C_a = 8.0$. The blowing factor is: $C_a = \epsilon h_c \mu/\mu_a = 0.45 \times 0.051 \times 10^{-3} \div 1.83 \times 10^{-5} = 1.25\ m^3/m^2$. Hence, the volume of blowing air is: $Va = 8 \times 1.25 = 10\ m^3/m^2$. Because ro $> 10^{11}\ m^{-2}$, there is no need to introduce a correction factor for turbulence.

9. We next determine the residual saturation, m_o, by first calculating the dimensionless capillary number:

$K_p = \Delta P/r_o h_c \sigma = 40{,}000/(2 \times 10^{12} \times 0.51 \times 0.04) = 9.8 \times 10^{-6}$, where mo = $aK_p^{-0.264}$. Hence, $m_o = 0.025(9.8 \times 10^{-6})^{-0.254} = 0.55$. In this expression a = 0.025.

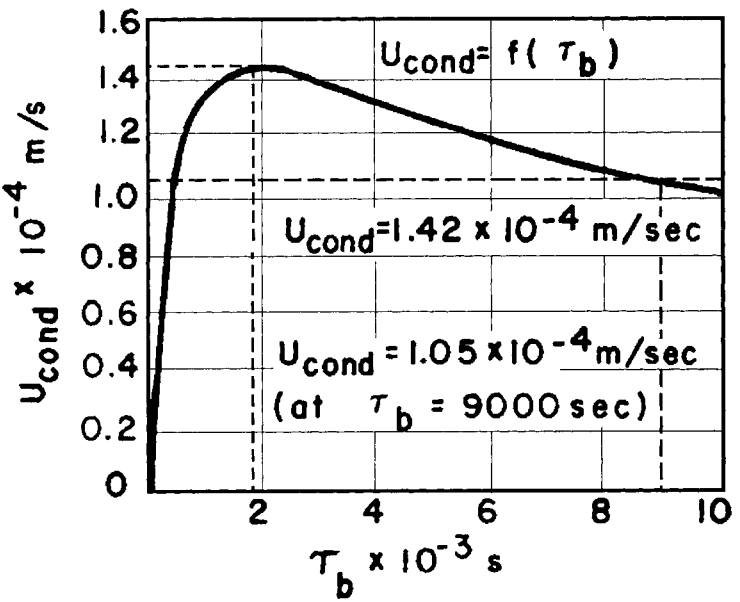

Figure 40. *Plot for Nutsche batch problem.*

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Parameter a is a constant that takes into consideration end-effects as the filtrate flows through the system. It is important to evaluate the decrease in average filter capacity per unit time for the total operating cycle if the total time of filtration, washing and drying changes from zero to a value several times larger than the maximum filter capacity. We trace the curve obtained by a plot of τ_b versus u_{cond}.The plot is illustrated in Figure 40. This is a three-step process, namely filtration, washing and drying, under conditions of maximum capacity. The curve shows a shallow slope towards the right from a maximum. This means that even for a large increase in the basic steps, for $\tau_b = \tau_{aux}$, no significant decrease in filter capacity is observed. Thus, the increase in τ_b from 1,800 to 9,000 s results in a decrease in filter capacity from 1.42 X 10^{-4} to 1.05 X 10^{-4} m^3/m^2-s, or a 1.35 times reduction.

Example for Optimum Capacity of Batch Filers at Constant Rate: We now consider a filter whose operating cycle consists of a filtration step at constant rate and auxiliary steps for filter preparation, which includes sludge feeding and cake removal. The cake anf filter plate are assumed to be incompressible, which leads to two common cases. In the first case we consider a constant filtration process whose filtration time changes along with filtrate volume and cake thickness. In this situation, $\tau_b = q/u$, where u denotes a constant rate of filtration. The average filtration rate is defined as:

$$u_{cond} = q/(\tau_b + \tau_{aux})$$

And substituting for τ_b into the above, we obtain:

$$u_{cond} = 1/[1/u + \tau_{aux}/q]$$

For $q \rightarrow 0$ or $\tau_b \rightarrow 0$, $U_{cond} \rightarrow 0$, and at $q \rightarrow \infty$ or $\tau_b \rightarrow \infty$, $U_{cond} \rightarrow u$. That is, to increase the filter capacity, filtration time should be increased. However, such an increase is limited by the maximum allowable pressure drop which at constant operating conditions, establishes a maximum cake thickness. Let's consider the following example for the first case in this analysis. We wish to determine the capacity of a batch filter operating at a constant rate. The rate of filtration is $q = 0.1 \times 10^{-3}$ m^3/m^2-s and the auxiliary time is 900 s. The solution to this problem is as follows. The filter capacity is characterized by the average conditional filtration rate, q_{cond}. Values of u_{cond} calculated are shown plotted in Figure 41. As shown, an increase in the amount of filtrate causes a sharp increase in filter capacity initially, with a limiting value of u attained eventually.

We now consider a second case, in which the filtration rate changes from one cycle to another; however, a constant rate is maintained during each cycle. The filtration is terminated when the pressure difference reaches a maximum allowable value. The amount of filtrate and cake thickness for each cycle will be different, as in Case 1, because the pressure difference depends not only on the cake thickness, but on the filtration rate as well. The following set of equations apply to this case:

$$u_{cond} = qu/(q + \tau_{aux}u)$$

$$q = \{\Delta P - \mu R_f u\}/\mu r_o x_o u$$

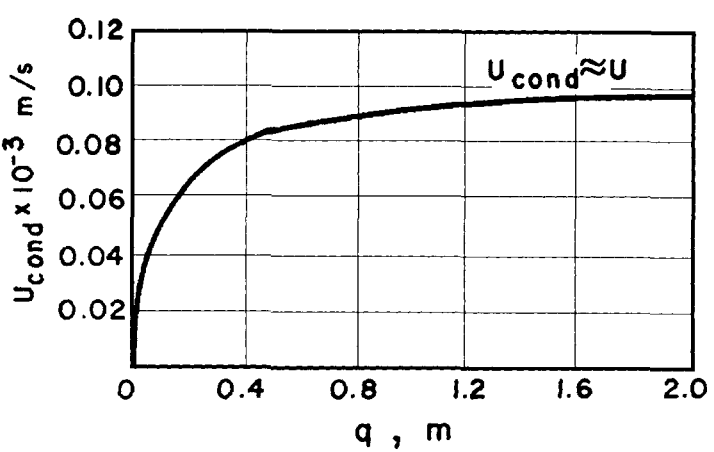

Figure 41. *Plot for example problem.*

In this model, ΔP is constant. This corresponds to the end of the filtration step.The following expressions apply:

$$u_{cond} = [\Delta Pu - \mu R_f u^2]/[\Delta P - \mu R_f u + \mu r_o x_o \tau_{aux} q^2]$$

$$u = \Delta P/[\mu R_f + (\mu r_o x_o \tau_{aux} \Delta P)^{1/2}]$$

The time of filtration corresponding to the maximum u_{cond} is:

$$\tau_b = \tau_{aux} + [(\mu R_f^2/\Delta P r_o x_o)\tau_{aux}]^{1/2}$$

To apply these equations, let's consider the following example. Determine a constant rate of filtration and the time of operation corresponding to the maximum capacity of a batch filter having the following conditions: maximum permissible pressure difference $\Delta P = 9\times10^4$ N/m^2; sludge viscosity $\mu = 10^{-3}$ N-s/m^2; filter plate resistance $R_f = 56\times10^{10}$ m^{-1}; specific cake resistance $r_o = 3 \times 10^{13}$ m^{-2}; $x_o = 0.333$; auxiliary time $\tau_{aux} = 600$ s; maximum permissible cake thickness $h_c = 0.025$ m. The solution is as follows:

The constant filtration rate is computed from the above formula:

$u = 9\times10^4/[10^{-3}\times56\times10^{10} + \{10^{-3}\times3\times10^{13}\times0.333\times600\times9\times10^4\}] = 0.0695\times10^{-3}$ m/s.

The filtration time is computed:

$\tau_b = 600 + [(10^{-3}(56\times10^{10})^2/9\times10^4\times3\times10^{13}\times0.333)\times600]^{1/2} = 1{,}056$ s.

The amount of filtrate is $q = \tau_b \times u = 0.0695\times10^{-3}\times1{,}056 = 0.073$ m.

Finally, the cake thickness is: $h_c = q\times x_o = 0.073\times0.333 = 0.025$ m.

The reader will find additional design calculation examples in the two literature citations provided in this section. in addition, refer to the Suggested Readings section of this chapter for further information.

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SEDIMENTATION EQUIPMENT

Thickeners and Clarifiers

There are a large number of processes in the chemical industries that handle a variety of suspensions of solid particles in liquids. The application of filtration techniques for the separation of these heterogeneous systems is sometimes very costly. If, however, the discrete phase of the suspension largely contains settleable particles, the separation can be effected by the operation of sedimentation. The process of sedimentation involves the removal of suspended solid particles from a liquid stream by gravitational settling. This unit operation is divided into *thickening,* i.e., increasing the concentration of the feed stream, and *clarification,* removal of solids from a relatively dilute stream.

Next Page

Chapter 7
MIXING EQUIPMENT

INTRODUCTION

Mixing is a unit operation that is practiced widely to meet a variety of process requirements. The specific mixing system design, operating arrangement and power requirements depend largely on the desired form of the intermediate or final products. Mixing is applied to achieve specified results in the following situations: creating a suspension of solid particles; the blending miscible liquids; dispersing gases through liquids; blending or dispersing immiscible liquids in each other; and promoting heat transfer between a fluid (liquid) and the coil or jacket of a heat exchanging device. The operating characteristics and design configuration of a mixing system are established on the basis of the required energy expenditure to create or approximate a homogeneous fluid system. For example, in producing an emulsion one must supply sufficient energy to "break up" the dispersed phase. In doing so, high shear stresses, which depend on velocity gradients, are developed in the mixing medium. In the zones in which the velocity gradient approaches a maximum, an intensive breaking up of the dispersed phase occurs. Mixing reduces concentration and temperature gradients in the processed system, thus exerting a favorable effect on the overall rates of mass and heat transfers. This applies in particular to dissolving applications, electrolysis, crystallization, absorption, extraction, heating or cooling, and heterogeneous chemical reactions, which proceed for the greater part in a liquid medium. Increased turbulence of the fluid system caused by mixing leads to a, decrease in the fluid's boundary layer thickness. This is derived from a continuous renewal of the surface contact area, resulting in a pronounced rate of increase in heat and mass transfer mechanisms. Regardless which medium is mixed with the liquid, i.e., gas, liquid or solid particles, two basic methods are employed. These are mechanical mixers, which utilize different types of impellers, and pneumatic mixers, which utilize air or an inert gas to effect mixing. In addition to these designs, mixing also is achieved in normal fluid handling operations, such as in pumps and jet flows.

Two major characteristics of all mixing devices that provide a basis for comparative evaluations are: (1) the efficiency of a mixing device, and (2) the intensity of mixing. The *efficiency* of a mixing device characterizes the quality of the process to be treated and may be expressed differently depending on the mixing purpose. For example, in producing suspensions mixing efficiency is characterized by the uniform distribution of the solid phase in the volume of equipment. For the intensification of thermal and diffusion processes, it is characterized by the ratio of

mixed and unmixed heat and mass transfer coefficients, respectively. Mixing efficiency depends not only on the equipment design, but also on the amount of energy introduced in the liquid being agitated.

The intensity of mixing is determined by the time required to achieve a desired technological result or by the mixer rpm at fixed process conditions (for mechanical mixers). From an economical standpoint, it is beneficial to achieve the required mixing effect in the shortest possible time. In evaluating the energy required for a mixing operation, one must account for the total energy consumption during the time needed to achieve a specified mixing result.

Mechanical mixers, which for the most part comprise rotating devices, are employed for liquids almost exclusively.

MECHANICAL MIXING EQUIPMENT

Mechanical mixing devices comprise three basic parts: an impeller, a shaft and a speed-reducing gearbox. The impeller constitutes the working element of the apparatus, mounted on a vertical, horizontal, or inclined shaft. The drive may be connected directly to a motor or through a gear box. The multitude of impeller configurations can be grouped into five distinct categories, of which only the first four are of major industrial importance. Discussions to follow concern only the three most widely used types, namely propeller, turbine and paddle mixers. The two basic propeller mixer configurations are fixed to a rotating vertical, horizontal or inclined shaft. The first is similar to an aircraft propeller, while the second resembles a marine propeller. Depending on the height of liquid layer, one shaft may carry one to three propellers. Due to their more streamlined shape, a propeller mixer's power requirements are less than the other types of mixers at the same Reynolds number. Their transition in the self-modeling region is observed at relatively low values of Reynolds number, Re $\approx 10^4$. They are capable of high-speed operation without the use of a gearbox and, hence, provide a more cost-effective operation because there are no mechanical losses in transmission. Propeller mixers produce an axial flow, which has a great pumping effect and provides short mixing times. Disadvantages compared to paddle and turbine mixers are higher cost, the sensitivity of operation to the vessel geometry and its location within the tank. As a general rule, propeller mixers are installed with convex bottom vessels. They should not be used in square tanks or in vessels with flat or concave bottoms.

A rotating propeller traces out a helix in the fluid, from which a full revolution moves the liquid longitudinally to a fixed distance, depending on its pitch, i.e., the ratio of this distance to the propeller diameter. Pitch may be computed from the following formula:

$$s = 2\pi r \tan(\psi)$$

where r is the propeller blade radius and, therefore, also the radius of the cylinder

created in the liquid as derived from the movement of the impeller, and ψ is the angle of tilt of the blade. Pumping and mixing efficiencies increase with pitch, as achieved by the axial flow of the liquid from the impeller. This flow results from the delivery head of the propeller and the helical turbulent flow of the entire contents of the vessel, which is caused by the radial velocity gradients in the liquid strata at different distances from the impeller. At high rotational speeds the entire fluid mass swirls despite the axial flow, and a central vortex begins to form around the shaft.

Draft tubes are employed to improve the mixing of large quantities of liquids by directing the motion of the liquid. Figure 1 shows such an arrangement, favorable for large ratios of liquid depth to mixer diameter. In such applications a high pumping capacity of the mixer is utilized, especially where mixtures of low viscosity are concerned. The draft tube directs the flow to the regions of the vessel that otherwise would not be agitated by the liquid stream. In the absence of draft tubes and at high rotational velocities of the propeller, baffles generally are located at various points in the vessel. Baffles minimize vortex formation and divide it into a number of local eddies, increasing the total turbulence of the tank. Depending on the application, multiple impellers may be mounted on a single revolving shaft and more than one shaft may be employed in a given tank. In some applications it is desirable to have two adjacent impellers rotating in opposite directions, forming a beater. Sometimes the impellers actually touch the walls of the tank, giving a positive scraping action, which is desirable when thick layers of material tend to stick to the wall.

Propeller mixers are used for mixing liquids with viscosities up to 2,000 cp. They are suitable for the formation of low-viscosity emulsions, for dissolving applications and for liquid-phase chemical reactions. For suspensions, the upper limit of particle size is 0.1 to 0.5 mm, with a maximum dry residue of 10%.

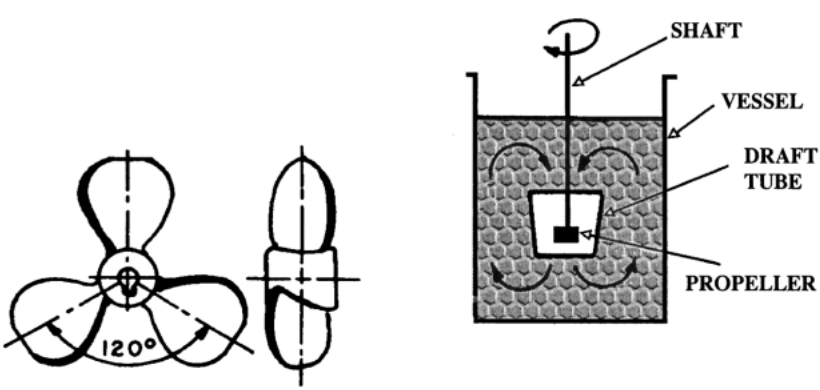

Figure 1. *Typical propeller mixer configuration.*

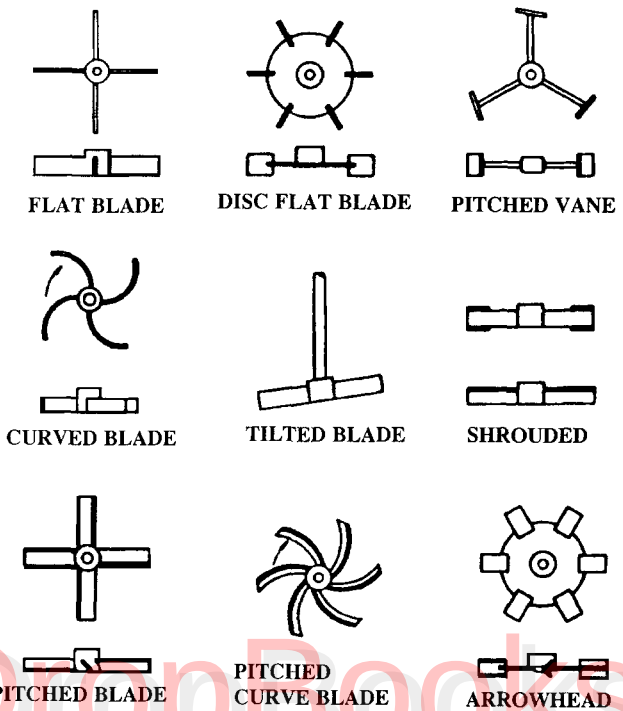

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Figure 2. *Turbine impeller configurations.*

Propeller mixers are unsuitable for suspending rapid settling substances and for the absorption of gases. Propellers are designed on the basis of data obtained from properly executed modeling experiments.

The turbine mixer represents a very important class. A turbine mixer as an impeller with essentially a constant blade angle with respect to a vertical plane, over its entire length or over finite sections, having blades either vertical or set at an angle less than 90° with the vertical. Blades may be curved or flat, as shown by the various configurations in Figure 2. Turbine mixer operation is analogous to that of a centrifugal pump working in a vessel against negligible back pressure. The mixing action is accomplished by the turbine blades, which entrain and discharge the liquid.

The predominantly radial flow from the impeller impinges onto the vessel walls, where it splits into two streams. These streams cause mixing by their energy. When turbine mixers are operated at sufficiently high rotational speeds, both radial and tangential flows become pronounced, along with vortex formation. This flow situation warrants the installation of baffles to ensure a more uniform flow distribution throughout the mixing vessel.

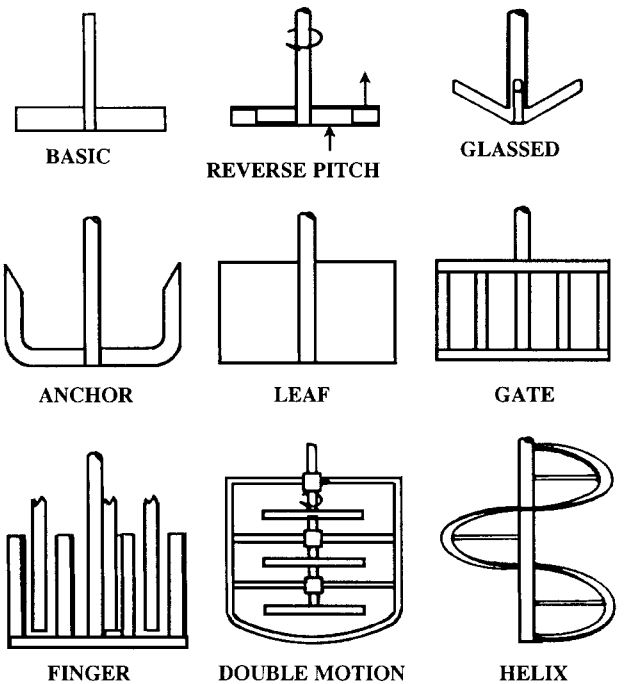

Figure 3. *Paddle mixer configurations.*

Paddle Mixers

Paddle mixers are devices consisting of two or more blades mounted on a vertical or inclined shaft. The basic paddle impeller configurations are shown in Figure 3. The main advantages of paddle mixers are their simplicity and low cost. A disadvantage is their small pumping capacity (a slow axial flow), which does not provide a thorough mixing of the tank volume. Perfect mixing is attained only in a relatively thin stratum of liquid in the immediate vicinity of the blades. The turbulence spreads outward very slowly and imperfectly into the entire contents of the tank; hence, circulation of the liquid is slow. Therefore, paddle mixers are used for liquids with viscosities only up to about 1,000 cp. Because of a concentration gradient that often is created in the liquid when these type mixers are used, they are unsuitable for continuous operation. This can be remedied by tilting the paddle blades 30 - 45° to the axis of the shaft, resulting in an increase in axial flow and, consequently, a decrease in concentration gradients. Such a mixer can maintain particles suspended, provided settling velocities are not high. Mixers with tilted blades are used for processing slow chemical reactions, which are not limited by

diffusion. To increase the turbulence of the medium in tanks with a large height to diameter ratio, a configuration is employed that consists of several paddles mounted one above the other on a single shaft. The separation between individual paddles lies in the range of 0.3 to 0.8 d (where d is the diameter of the paddle) and is selected according to the viscosity of the mixture. For mixing liquids with viscosities up to 1,000 cp, as well as for heated tanks in cases in which sedimentation can occur, anchor or gate paddle mixers are employed (refer to Figure 3). Paddle diameters are almost as large as the inside diameter of the tank in such applications, so that the outer and bottom edges of the paddle scrape (or clean) the walls and bottom.

Leaf-shaped (broad blade) paddle mixers provide a predominant tangential flow of liquid, but there is also turbulence at the upper and lower edges of the blade. Leaf-type blades are employed for mixing low-viscosity liquids, intensifying heat transfer processes, promoting chemical reactions in a reactor vessel and for dissolving materials. For dissolving applications, leaf blades usually are perforated. During the mixer's rotation, jets are formed at the exits from the holes which promotes the dissolution of materials. The rotational velocity of paddle mixers is in the range of 15 to 45 rpm. Under these conditions. the pumping action is small and there generally is no danger of vortex formation. As such, paddle mixers are most often used in vessels without baffles. However, for broad-blade paddles, which operate at speeds up to 120 rpm, baffles are incorporated into the design to minimize vortex formation.

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Emulsifiers, Blenders and Planetary Mixers

There are a number of other mixing configurations that have widespread applications to both liquid-solid and solid-solid blending operations. Emulsifiers are used as an alternative to slow-speed impeller mixing or high-pressure homogenization for a wide range of processing requirements. Typical applications include the preparation of adhesives (e.g., asphalts, carbon dispersions, clay dispersions, dyestuffs, paints and inks, lacquers), cosmetics (e.g., creams, emulsions, hand lotions, perfumes, shampoos, deodorants), foods (e.g., chocolate coatings, mustard, soft drinks, sugar emulsions), pharmaceuticals (e.g., antibiotics, ointments, reducing animal tissues), plastics (e.g., cold cutting resins, polyester dispersions, resin solutions) and various miscellaneous mixtures such as floor polishes, gum dispersions, lubricants, petroleum emulsions, etc. These types of mixers normally are used in dished or conical bottom vessels. The mixing process can be thought of as performed in three stages. In the first stage, the high-speed rotor operating at close clearance to the stator draws material in from the bottom of the mixing vessel and subjects it to intense mixing. In stage 2, the rotor accelerates the product toward the blade periphery. There it is expelled through openings in the stator into the body of the mix while undergoing an intensive mechanical and hydraulic shearing action. At the same time, new material is drawn

into the center of the rotor. In the third stage, the expelled mixture is deflected by the tank wall, completing the circulation. To increase circulation or to create a vortex for the incorporation of light solids, a downthrust propeller may be mounted on the rotor shaft.

The operating sequence for emulsifiers varies with the intended application. Standard batch units are available through 100 hp. For most applications the rotor head should be mounted two to three head diameters above the vessel floor. Obviously, with these designs a slender-shaped vessel generally is preferred. Maximum mixing efficiency generally is achieved on mixtures under 10,000 cp. The principal criterion for achieving the best efficiency is to maintain a maximum circulation of all materials through the rotor/stator at all times during the process cycle. As viscosity increases, flow through the head decreases, thus lessening the work on a given volume of material while it is being circulated within the vessel. Inline emulsifiers are less susceptible to this problem as they may be fed by means of positive displacement pumps. Note that because of the wide range of applications to which a single design is subjected, the density and rheology of the mixture are variables. The pumping capabilities of a unit on materials that have the same apparent viscosity may not be the same in actual practice. For example, many polymers have a low apparent viscosity and appear ideally suited to a particular unit design and application criteria. However, on testing, it may be observed that the flow is poor and that the end result is only marginal. Often, modified tank configurations, auxiliary agitation, different rotor speeds or head diameters will improve efficiency. Pilot testing almost always is recommended for accurate scaleup.

Ribbon blenders resemble helical paddle mixers both in design and operation, the principal difference being that the shaft orientation is strictly horizontal. There are basically three standard agitator designs, namely, continuous ribbon, interrupted ribbon, and paddle type. They may be arranged for either center or end discharge from the mixing vessel. The principal designs are shown in Figure 4. The continuous ribbon, arranged for center discharge, produces homogeneous blends relatively quickly. The outer ribbons move materials toward the center, whereas inner ribbons move materials toward end plates. Both provide radial movement of materials. In the case of continuous ribbons, the outer ribbons move materials towards the discharge end of the vessel. The inner ribbons move materials in the opposite direction. Both provide radial transfer of materials. The interrupted ribbon provides the same basic action as a continuous ribbon agitator. It requires less power, and generally is applied to materials with high bulk densities. The paddle-type is often used in such applications as in the preparation of instant beverage premixes.

Ribbon blenders are essentially self-contained mixers. They are employed in a variety of solid-liquid, solid-solid, and liquid-liquid blending applications in the chemical process industries. Examples include plastics, pigments, pharmaceuticals, specialty chemicals, confectionary.

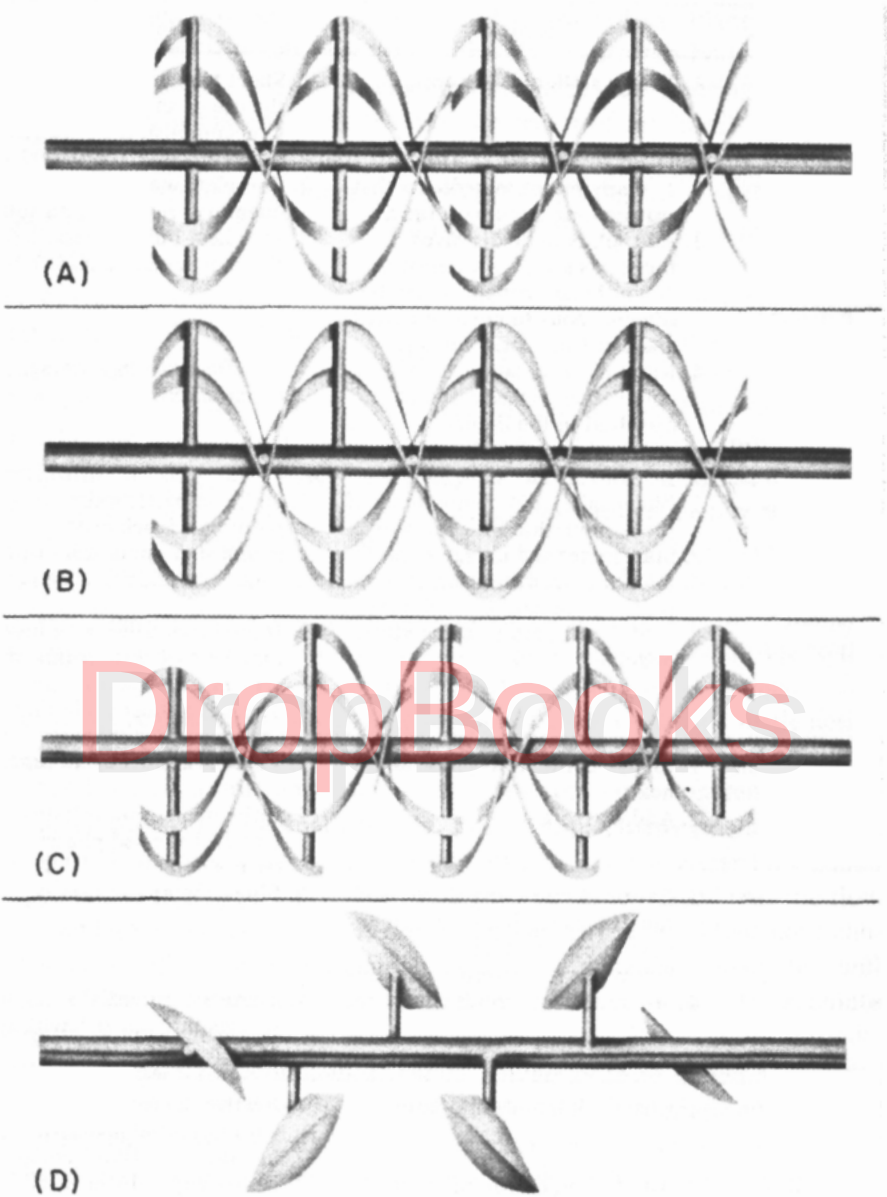

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Figure 4. *Ribbon blender agitator configurations:(A) continuous ribbon, arranged for center discharge; (B) continuous ribbon arranged for end discharge; (C) interrupted ribbon; (D) paddle-type ribbon.*

The reader may consult the reference section of this chapter for citations dealing with these types of agitator configurations and specific applications.

Planetary mixers are used for a variety of liquid and solid mixing applications, from simple mixtures to sophisticated reactions, involving high temperature, vacuum or internal pressure. This type of mixer is employed in batch operations. During the mix cycle, two rectangular-shaped stirrer blades revolve around the tank on a central axis. Each blade revolves on its own axis simultaneously, at approximately the speed of the central rotation. With each revolution on its own axis, each stirrer blade advances forward along the tank wall. This movement provides homogeneity of the material being mixed and does not depend on the flow characteristics of the mix. Instead, the stirrers cover every point within the mix tank. Double planetary mixers have no packing glands or bearings in the product zone. Hence, cleaning between batches is minimized. Figure 5 shows a planetary mixer unit. The machine is equipped with a hydraulic lift that permits the stirrer blades to be lowered and raised in and out of the mixing tank. The unit shown is capable of handling low-viscosity fluids to very high- viscosity pastes and doughlike materials. Typical capacities of these units range from 10 to more than 300 gallons.

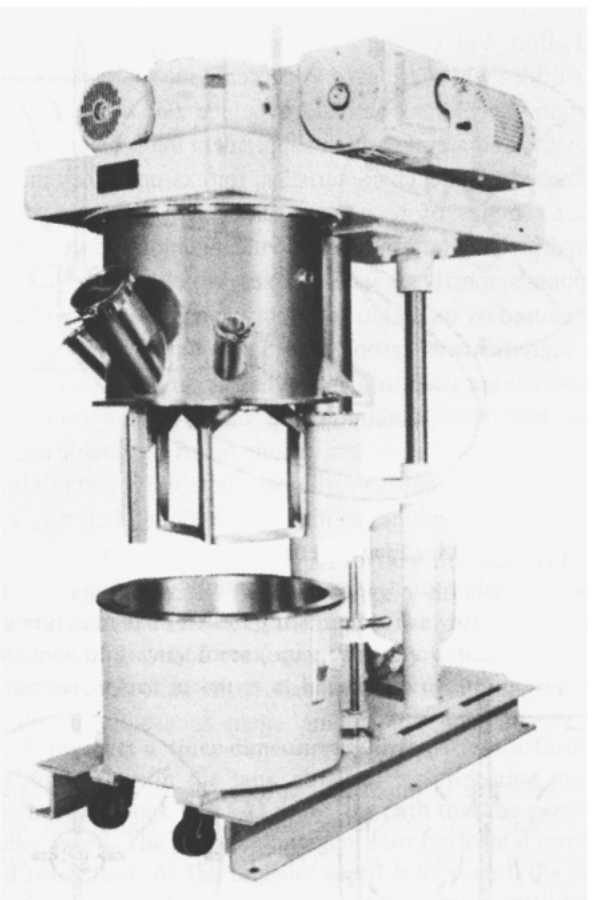

Figure 5. *Planetary mixer with a vacuum hood.*

Rubber Compounding Equipment

Synthetic rubber or elastomers, in its raw state is too plastic for most commercial applications. Through a curing process termed vulcanizing, raw rubber can be made to lose plasticity and gain elasticity. By compounding the raw or "neat" rubber with various types and amounts of additives before the vulcanizing, tensile strength, abrasion resistance, resiliency, heat aging, and other desirable properties can be imparted to the rubber. The proportions and types of additives (including vulcanizing agents) compounded into the raw rubber, and the vulcanizing temperature, pressure, and time are varied in accordance with the properties desired in the final product. After the rubber is compounded, it is formed into the desired shape and then cured at the required temperature. In the forming steps, large amounts of organic solvents are often used in the form of rubber adhesives. Types of additives that are compounded into the rubber may be classified as vulcanizing agents, vulcanizing accelerators, accelerator activators, retarders, antioxidants, pigments, plasticizers and softeners, and fillers. Vulcanizing agents include peroxides and sulfur. Vulcanizing accelerators include aldehyde-amines, guanidines, and thiuram sulfides which are used to decrease the time and temperature required for vulcanization. Accelerator activators include zinc oxide, stearic acid, litharge, magnesium oxide, and amines which supplement the accelerators and, in addition, modify the finished product characteristics; for example, they increase the modules of elasticity. Examples of retarders include salicylic acid, benzoic acid, and phathalic anhydride to retard the rate of vulcanization. Antioxidants include many organic compounds, mostly alkylated amines, which are used to retard deterioration of the rubber caused by oxidation and improve aging and flexing ability. Pigments include such ingredients as carbon black, zinc oxide, magnesium carbonate, and certain clays which are used to increase tensile strength, abrasion resistance, and tear resistance. Iron oxide, titanium oxide, and organic dyestuffs are used to color the rubber. Plasticizers and softeners include resins, vegetable and mineral oils, and waxes which are used to improve resiliency, flexibility, and mixing and processing characteristics. Fillers include whiting, slate flour, barytes, and some of the pigments previously mentioned are used to improve processing properties and lower the cost of the finished product. In the compounding of blends, the accelerators are added first to the mass of raw rubber being milled or mixed. Then a portion of the plasticizers (if present in the blend recipe) are added, followed by the reinforcing pigments, the remainder of the plasticizers, the antioxidants, and any inert fillers or coloring agents. The vulcanizing agent is usually introduced as the last ingredient.

In order to be effective in imparting various chosen characteristics, all additives employed in a blend must be homogeneously dispersed throughout the blend. The two most commonly employed pieces of equipment for blending rubber and additives are rubber mills and a special type of mechanical mixer known as the Banbury mixer. A typical rubber mill consists of two rolls which rotate toward each

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other at different speeds , creating a shearing and mixing action. Neat rubber is placed in the mill, and the additives are introduced, generally one or two components at a time. Additives may be finely divided solids or liquids. Another device commonly used for compounding rubber stock is the Banbury mixer.

Figure 6 shows cross-sections of two typical Banbury mixers. Each consists of a completely enclosed mixing chamber in which two spiral-shaped rotors, revolving in opposite directions and at different speeds, operate to keep the stock in constant circulation. A ridge between the two cylindrical chamber sections forces intermixing, and the close tolerances of the rotors with the chamber walls results in a shearing action. A floating weight in the feed neck confines the batch within the sphere of mixing. This combination of forces produces an ideally homogeneous batch.

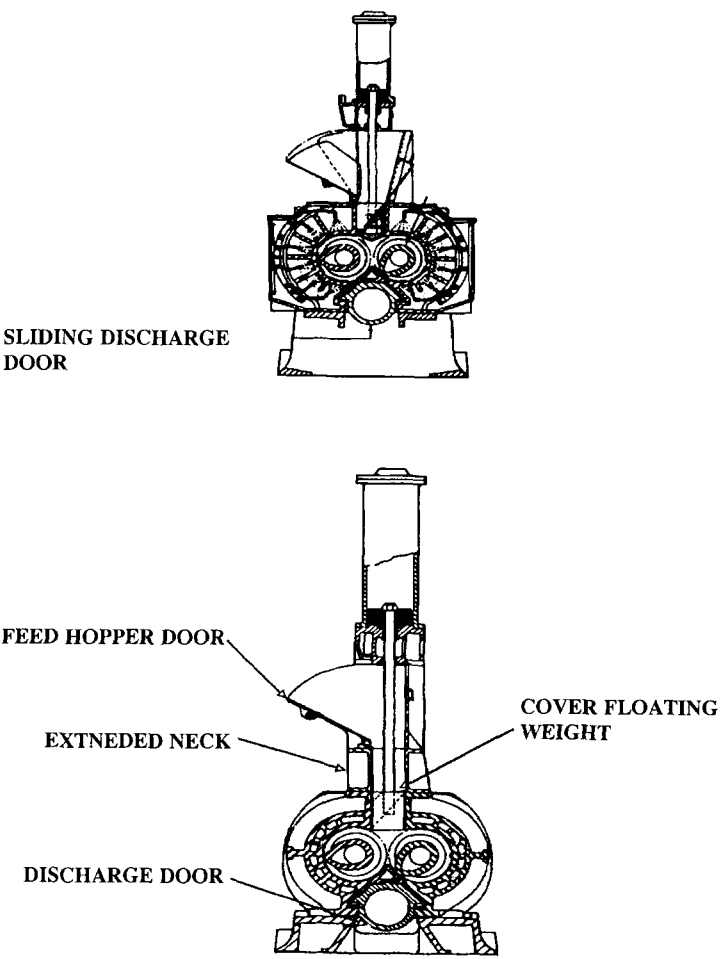

Figure 6. *Banbury mixer units.*

Flow Patterns in Agitated Tanks

The forces applied by an impeller to the material contained in a vessel produce characteristic flow patterns that depend on the impeller geometry, properties of the fluid, and the relative sizes and proportions of the tank, baffles and impeller. There are three principal types of flow patterns: tangential, radial and axial. Tangential flow is observed when the liquid flows parallel to the path described by the mixer as illustrated in Figure 7.

When the flow pattern in a mixed tank is primarily tangential, the fluid discharge from the impeller to the surroundings and its entrainment into the impeller are small. Also, fluid transfer in the vertical direction is at a minimum. The mixing effect is lowest when the rotational velocity of the liquid approaches that of the mixer.

With radial flow, the liquid discharges from the impeller at right angles to the mixer axis and moves along the radius. As soon as the centrifugal force, which depends on the impeller's diameter and speed, overcomes the resistance of the medium, it initiates radial flow from the impeller into the bulk fluid. Figure 8 shows the flow pattern of a mixer producing radial flow in two sections of the tank. In the bottom part of the tank, the impeller entrains the liquid in an upward direction and displaces it at right angles to the axis of the impeller. In the upper part of the tank the impeller entrains the liquid downward, also displacing it perpendicular to the impeller axis.

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Axial flow, in which the liquid enters the impeller and discharges along a parallel path to the axis, is shown in Figure 9. The radial and longitudinal components are primarily responsible for the derived mixing action. The tangential component is important when the shaft has a vertical orientation and is positioned near the center of the tank.

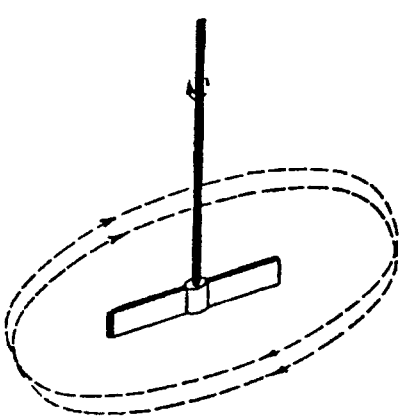

Figure 7. *Tangential flow pattern generated by a paddle mixer.*

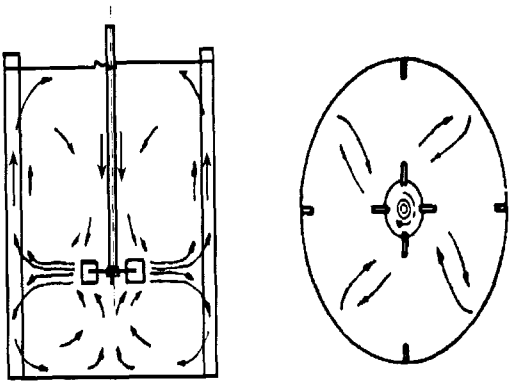

Figure 8. *Axial flow patterns in a tank.*

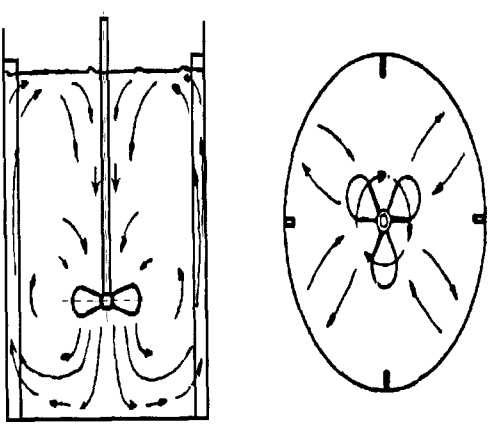

Figure 9. *Axial flow patterns.*

Based on the pitch of the impeller with regard to the direction of rotation, there are two possible axial flow patterns: that in which the impeller pumps the liquid from the bottom to the surface; and that in which the impeller pumps liquid from the surface to the bottom. A combination of the three principal types of flow normally is encountered in mixing tanks. The tangential flow following a circular path around the shaft forms a vortex at the surface of the liquid. The vortex formation results from the influence of gravity forces, quantitatively determined by means of the Froude number, which increases at higher speeds, promoting vortex formation. Figure10 presents a three-dimensional flow pattern, affording a clear image of the liquid flow in the tank obtained by projecting the path of a liquid particle in two planes. Part (A) shows the path that the particle takes at a given impeller speed.

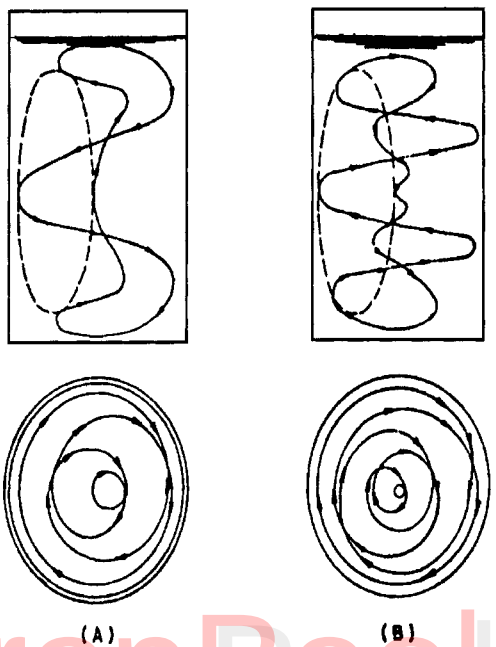

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Figure 10. *Three-dimensional flow pattern.*

The particle undergoes four horizontal revolutions per single vertical revolution. As the impeller speed is increased, the particle performs a greater number of horizontal revolutions per one vertical revolution (Figure 10 (B)), showing that the greater centrifugal force at higher speeds increases the radius of curvature of the paths of the particle, which thus is shifted farther from the center of the tank. Further increases of the rotational speed and, subsequently, the centrifugal force, result in higher ratios between the horizontal and vertical revolutions of the particle and, thus, larger radii of curvature for particle paths. A particle subjected to gravitation also is acted on by tangential acceleration, as determined by the centrifugal force developed by the impeller, and influenced by the density and viscosity of the mixture. The direction of resultant acceleration is perpendicular to the surface of the liquid at any given point. If the tangential acceleration is negligible, the level of the fluid will be horizontal. Otherwise, the action of the centrifugal field becomes pronounced and the liquid level exhibits a depression that is referred to as the vortex.

Vortex formation is a condition that arises from centrifugal acceleration acting on gravitational acceleration. The circular motion of the entire contents of the tank predominates over the flow of the liquid from the impeller. Flow orientation thus is important not only in cases of noticeable vortex formation, but

also in mixers with tangential flow. In the latter case, at certain mixer dimensions and impeller speeds the circulating liquid in the tank may attain the same velocity as the impeller. Mixing becomes ineffective under such conditions.

Vortex formation leads to a considerable drop in mixing efficiency and should be suppressed as much as possible in practical applications to increase the homogenizing effects of mixers. The preferable method of vortex suppression is to install vertical baffles at the walls of the mixing tank. These impede rotational flow without interfering with the radial or longitudinal flow. Figure 11 illustrates such a system.

The distribution of velocity components (radial, tangential and axial) under conditions of mixing with baffles in comparison with the conditions of vortex formation is presented in Figure 12. The dashed lines in Figure 12 indicate non-baffled conditions. Comparison of the non-baffled and fully baffled velocity curves (solid line) leads to the following set of conclusions on vortex suppression when dealing with perfectly miscible liquids:

- Baffles are responsible for restricting the tangential velocity component, u, and augment the vertical component, while simultaneously increasing the radial velocity, U_r. The net result is that the liquid discharges from the impeller in a wider flow radius.
- The streamlined distribution becomes more regular when baffles are used. Hence, the energy transmitted from the impeller to the liquid is utilized more uniformly.
- The circulation increases, and the difference between the circulation rate under fully baffled conditions and at the vortex formation rises to two to four times the original value. This means that the power input increases considerably in the range of two to ten times the input without baffles.

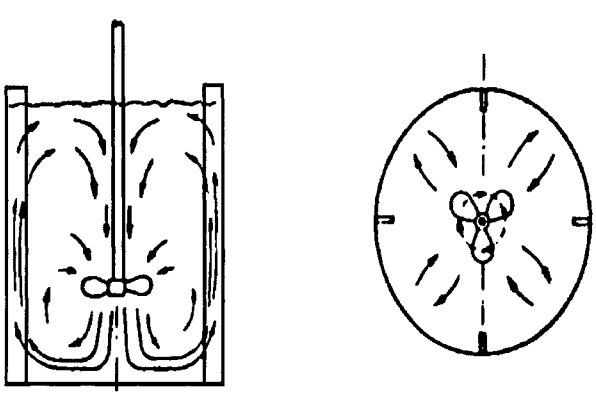

Figure 11. *Flow patterns in a baffled tank with centrally mounted impeller.*

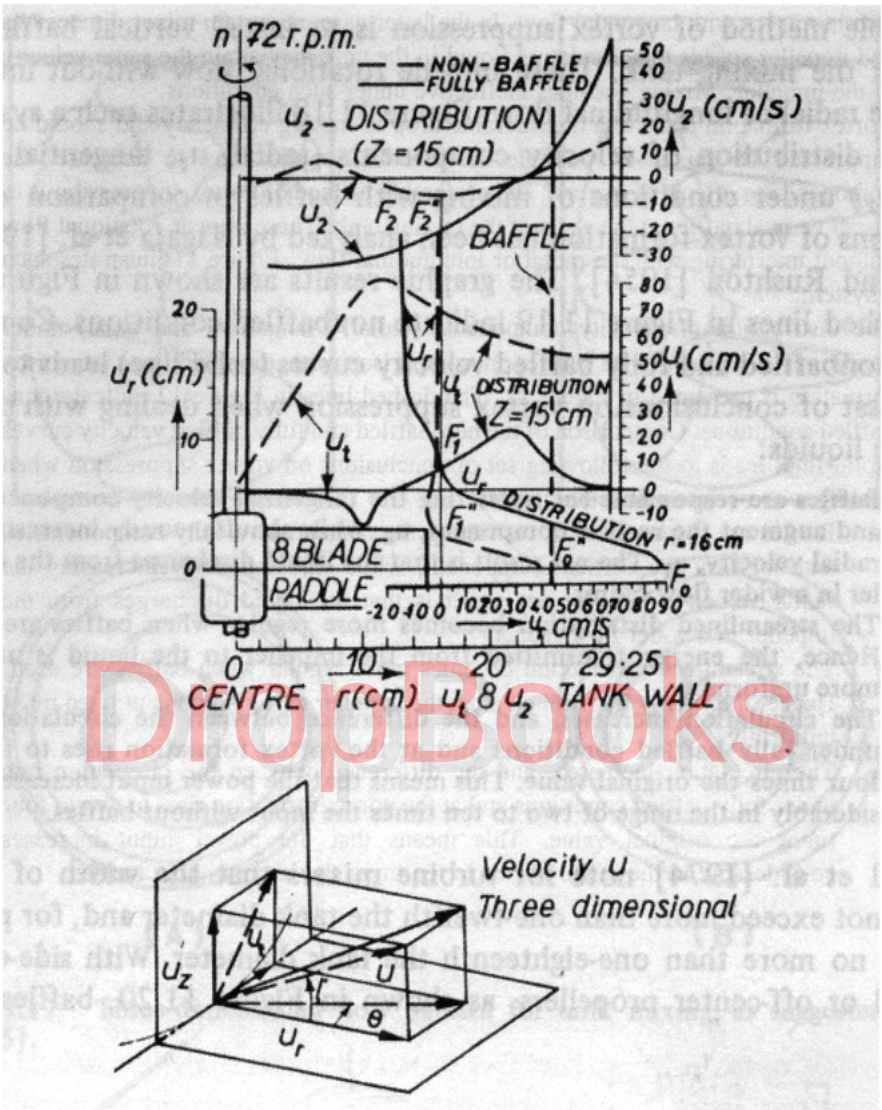

Figure 12. *Variation of velocity distribution in a mixing tank on insertion of full side wall baffles.*

For turbine mixers that the width of a baffle should not exceed more than one-twelfth of the tank diameter and, for propeller mixers, no more than one-eighteenth the tank diameter. With side-entering, inclined or off-center propellers, as shown in Figure 13, baffles are not required. Instead, shrouded impellers and diffuser rings may be used to suppress vortex formation. These devices contribute to flow resistance and reduce circulation by creating intense shear and abnormal turbulence

at the impeller discharge. The specific flow pattern in the tank depends on the type of impeller if vortex formation and swirling are eliminated. With propeller mixers, the liquid is driven straight down to the bottom of the tank, where the stream spreads radially in all directions toward the wall, flows upward along the wall, and returns to the suction of the propeller from the top. Paddle mixers provide good radial flow in the immediate plane of the impeller blades but are poor in developing vertical currents. Turbine mixers drive the liquid radially against the wall, where the stream divides into two portions. One portion flows downward to the bottom and back to the center of the impeller from below; the other flows upward toward the surface and back to the impeller from above. Figure 14 shows that there are two separate circulation currents generated.

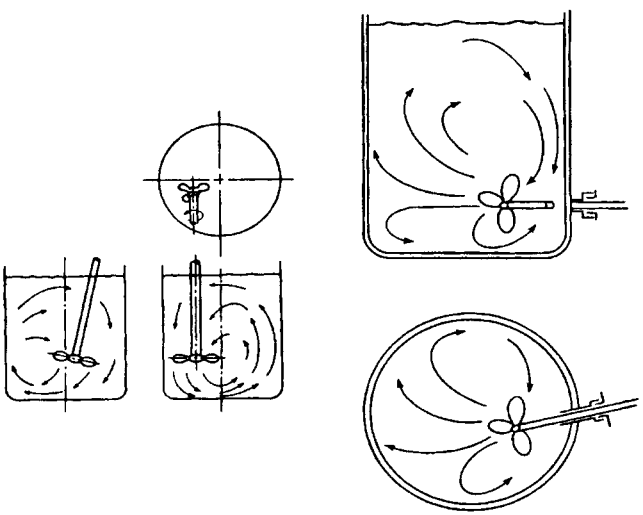

Figure 13. *Off-center impeller flow patterns.*

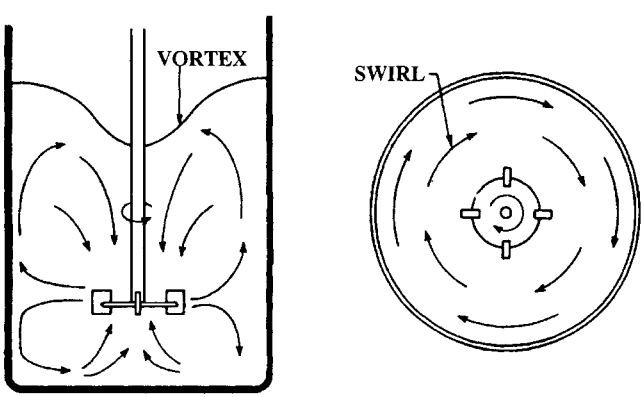

Figure 14. *Circulation patterns in an agitated tank.*

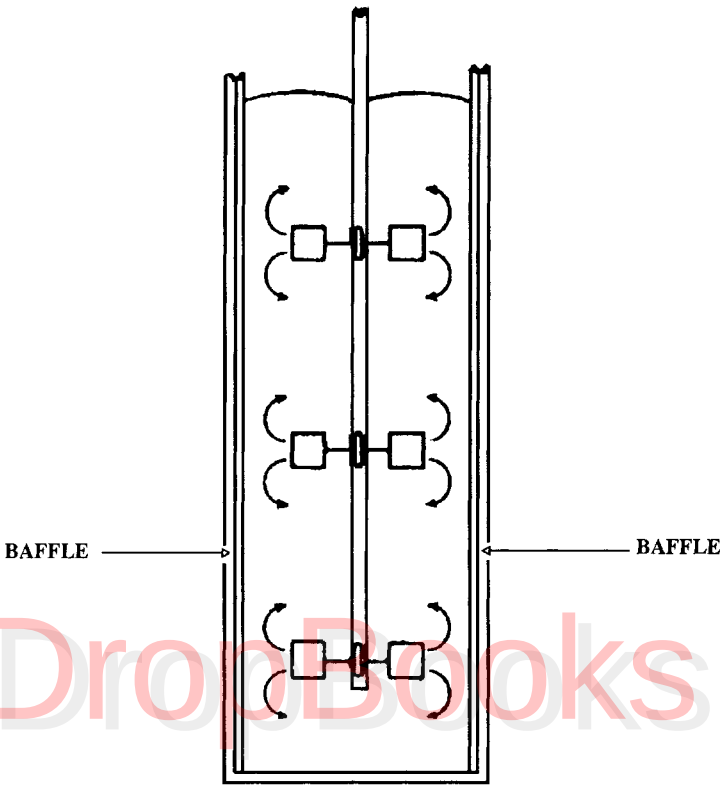

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Figure 15. *Multiple turbines in a tank.*

Turbines are especially effective in developing radial currents, but they also induce vertical flows, especially under baffled conditions. In vertical cylindrical vessels, the ideal liquid depth for good mixing should be somewhat greater than the tank diameter. If greater depths are required, two or more impellers may be installed on the same shaft, with each impeller serving as a separate mixer. Two circulation currents are generated for each mixer, as shown in Figure 15 .

The bottom impeller should be mounted about one impeller diameter above the bottom of the tank. As noted earlier, when the direction and velocity of the flow to the impeller suction are to be controlled, draft tubes are employed. These devices are designed to set in motion regions in the tank that otherwise would not be agitated. Typical flow profiles are shown in Figure 16. Draft tubes are mostly used with propellers and really amount to mixing by circulation of the contents through a thoroughly agitated region in the vicinity of the impeller. The turbulence induced in the tank by the flow of liquid from the draft tube only contributes to the blending action.

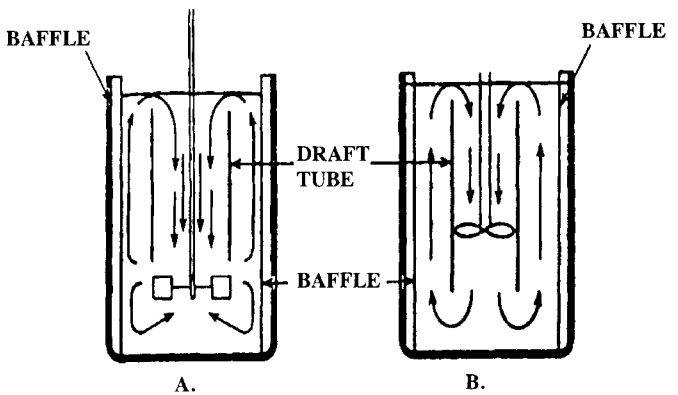

Figure 16. *Draft tubes in baffled tanks: (A) turbine; (B) propeller.*

DESIGN PRACTICES

In the previous section, a description of the various types of equipment commonly used for carrying out mixing operations was provided. Emphasis was placed on mechanical agitators since these have the broadest range of application, however there are other important devices worth noting, such as jet, pump, eductor, orifice, and inline mixing devices. Some of these important pieces of equipment are briefly described below.

Specifying a mixing device is a two-stage process. Mixing needs of the application first have to be quantified and then these needs have to be translated to an equipment design. Included in this section is the translation from mixing need to specified equipment design. Principles addressing how to quantify the mixing need for systems of the common phase configurations of miscible fluids, gas-liquid, immiscible liquids, and solid-liquid are reviewed.

As noted earlier, mixing operations are carried out to blend two or more miscible fluids, to suspend solids in a liquid to form a slurry (which may then dissolve or react) , to disperse a gas into a liquid to carry out mass transfer, and to disperse one immiscible liquid into another for mass transfer or to form an emulsion. For any of these operations, equipment sizing and specification can be based upon a numerical description of the mixing requirement. This mixing equipment can be expressed as a given discharge flow rate from the mixing device, a given total mixing power or power per unit volume dissipation by the mixing device, a given bulk fluid velocity, or in the case of turbine agitators a given tip speed. Before addressing this in more detail, some basic terminology needs to be introduced:

Agitator: a general term used to describe a device which imparts motion to a fluid.

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Chapter 8
CALCULATIONS FOR SELECT OPERATIONS

INTRODUCTION

This chapter provides a compendium of short engineering calculation methods and formulas for select process operations and equipment. The formulas and procedures provided in this chapter offer simple, rapid estimates of key parameters important to specifying and obtaining various engineering parameters. The information is derived from the open literature, and is believed to be accurate for obtaining better than an order of magnitude estimate for each calculation.

HEAT CAPACITY RATIOS FOR REAL GASES

For many calculations it is required to know the ratio of C_P/C_v for a gaseous compound. The well-known expression C_p-C_v = R (where R is the universal gas constant) holds good only for gases under ideal conditions. When working with situations involving real gases, the relationship between C_P and C_v is given as:

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$$C_p - C_v = R\,(1 + Pr / T_r^2\,(0.132 + 0.712/T_r)^2)$$

where: P_r = Reduced pressure = P/P_c
T_r = Reduced temperature = T/T_c
P_c = Critical pressure
T_c = Critical temperature in absolute units

For C_P and C_v in metric units, the value for R is 1.987 cal (gm-mole)-°K

SIZING OF VAPOR-LIQUID SEPARATORS

Vapor-liquid separators (drums) are used to separate a liquid from a vapor-liquid stream with a minimum of liquid carryover. The separator size is determined by the vapor velocity which depends on the entrainment method used. The working equation is:

$$D = [(\text{Vapor Load})/0.178\,R_d]^{1/2}$$

where vapor load = cfs of vapor $[\rho_v/(\rho_L-\rho_v)]^{0.25}$. ρ_L and ρ_v are the liquid and vapor densities, respectively, and R_d is a ratio that varies with application. The various values of R_d are summarized below:

Operating Condition	R_d
Usual process application with mesh	1.54
Pressure surges (with mesh)	1.15
Usual process application without mesh	0.2 - 0.44
Knock out drum	0.88

OVERALL EFFICIENCY OF A COMBINATION BOILER

A combination boiler uses more than one fuel, at any given time. A good example is a power boiler in a pulp and paper mill using coal, wood waste, natural gas, and fuel oil. A common calculation method for a boiler burning bark (at 55% moisture), natural gas, and using Bunker C fuel oil is outlined. The method involves one simple step to include a term for coal if needed. Also, the method for computing efficiency is different from the standard methods that involve combustion calculations. The overall method is simpler to use and therefore more appealing to power plant personnel. This simplicity is important to an energy management program because boiler efficiencies should be tracked on a daily basis. We first define the Btu input into the boiler = Btu in fuels + Btu in feedwater; and Btu output from the boiler = Btu in steam out. The overall efficiency is then defined as Btu in steam out/Btu in fuel in and feedwater in. Hence:

$$\text{Eff} = S \times h_s/\{S \times h_w + (\sigma \times 150{,}000) + (g \times 10^3) + (W \times 2000 \times 4000)\}$$

where S = lb/hr steam generated
h_s = enthalpy of steam out, Btu/lb
h_w = enthalpy of feedwater in, Btu/lb
σ = gallons of Bunker C fuel oil/hr
W = tons/hr of bark usage

PUMP HORSEPOWER CALCULATIONS

A rapid method to estimate pump horsepower is by the following formula:

$$HP = GPM\ (\Delta P)/1715(\eta)$$

where HP = Pump horsepower
GPM = Gallons per minute of liquid being pumped
ΔP = Delivered pressure (discharge minus suction), psig
η = Pump efficiency

Note that the expression can be rearranged to obtain ΔP in terms of other variables. The expression can therefore be used to generate a family of curves of HP versus ΔP with GPM as a parametric variable.

PUMP EFFICIENCY CALCULATIONS

An approximate method to estimate pump efficiencies is based on the following formula. The expression is valid for flows between 100 and 1,000 gpm, and for developed heads between 50 and 300 ft.

$$\eta = 80 - 0.2855F + 3.78\times10^{-4}FG - 2.38\times10^{-7}FG^2 + 5.39\times10^{-4}F^2 - 6.39\times10^{-7}F^2G + 4\times10^{-10}F^2G^2$$

where η = Pump efficiency expressed as percent
F = Developed head, ft
G = Flowrate, gpm

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LIME KILN PRECOAT FILTER ESTIMATION

A lime kiln is a common piece of equipment used in the chemical process industry for decomposing calcium carbonate into calcium oxide and carbon dioxide. In the pulp and paper industry the slurry, which consists of a mixture of calcium carbonate, water and some inert materials, is filtered to remove a portion of the water. The filter that is employed is a rotary drum filter, commonly called the precoat filter. The fuel consumption in the kiln depends upon the amount of water entering the kiln. A useful calculation that is often made is aimed at determining the fuel savings when the mud solids to the kiln are increased incrementally. The calculation method outlined below is derived on the basis of one minute of operation, and using the mass balance diagram given in Figure 1.
For one minute of operating time, we may state that:

$$\text{Mass of slurry in} = (X \text{ gal})(62.4 \text{ lm/ft}^3)(1\text{ft}^3/7.43 \text{ gal}) = 8.34\ X, \text{ lb}$$
$$\text{Mass of solids in} = 8.34\ SX\eta/100, \text{ lb}$$

In this expression, X is the mud flowrate to the precoat filter, η is the percent solids in the mud to the precoat filter, and S is the specific gravity of the mud.

$$\text{Mass of water in} = 8.34\ SX\ [(100-\eta)/100]$$

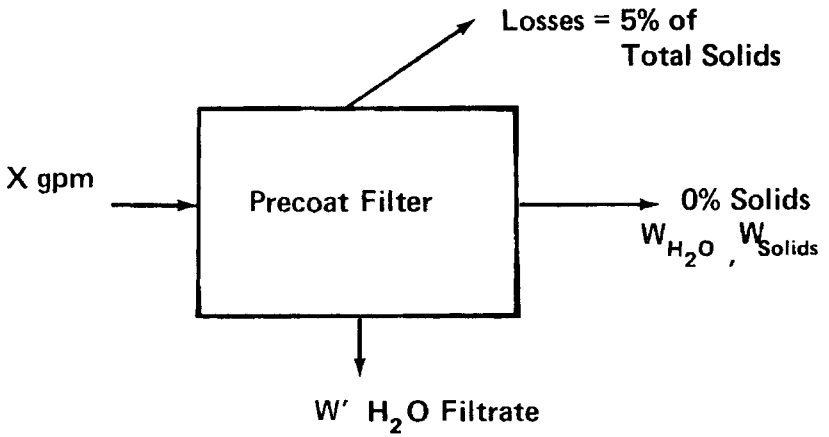

Figure 1. *Material balance diagram for fuel savings in a lime kiln.*

Applying a water balance, we obtain:

$$(8.34)(SX)[1-\eta/100] = W_{H2O} + W'_{H2O}$$

Applying a solids balance, we obtain:

$$(0.95)(8.34SX)\eta/100 = W_{solids}$$

Defining ϕ as the percent solids in the mud from the precoat filter, then:

$$W_{solids}/(W_{H2O} + W_{solids}) = \phi/100$$

Combining the last two expressions, we obtain:

$$\{(0.95)(8.34)SX\eta/100\}/\{W_{H2O} + (0.95)(8.34SX\eta/100)\} = \phi/100$$

and solving for W_{H2O}, we obtain:

$$W_{H2O} = \{(0.95)(8.34)(Sx\eta)/\phi\}\{1-\phi/100\}$$

The Btu needed to evaporate this quantity of water and to raise the vapor to the back-end temperature of the kiln is:

$$Btu/min = (W_{H2O})(1,1110) + W_{H2O}\,(t_b - 212)(0.473)$$

where t_b is the temperature in the feed end of the kiln in °F. On a per year basis, the Btu quantity is:

$$\text{Btu/yr} = (\text{Btu/min})\ (1{,}440\ \text{min/day})\ (\text{N days operation/yr})$$

Hence, the gallons of fuel oil required for the operation are:

$$G = \text{Btu per year}/(150{,}000 \times 0.85)$$

Combining the above expressions, we obtain:

$$G = 0.08948\ NSX\eta/\phi[1-\phi/100][1{,}110 + 0.473(t_b - 212)]$$

The decrease in fuel oil consumption when the solids out decreases from ϕ_1 to ϕ_2 is therefore given by the following expression:

$$\Delta G = 0.08948\ NSX\eta[(1 - \phi_1/100)\ 1/\phi_1 - (1 - \phi_2/100)\ 1/\phi_2] \times [1{,}110 + 0.473(t_b - 212)]$$

This last expression serves as the working formula. Note that natural gas may be substituted for fuel oil, whereby the only resulting change will be the heating value in the derivation.

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STEAM SAVINGS IN MULTIPLE EFFECT EVAPORATORS

Multiple-effect evaporators are widely used in the chemical process industry for concentrating dilute solutions of water and a recoverable solute (refer to Chapter 3). A measure of effectiveness of the evaporator set is the steam economy which is expressed in lbs of water evaporated per lb of steam used. Upgrading existing sets or adding new sets affects the steam economy. Any improvement (increase) in the steam economy is beneficial. The question is whether or not the resulting savings will yield an attractive ROI (Return on Investment). The following calculation estimates the change in steam consumption when the steam economy is increased incrementally. The expression derived is applicable to any multiple effect evaporator set. The inputs required are flows, concentrations, density of the dilute solution, and the steam economies. Assuming a basis of one hour of operation, then:

lbs solution into system = $60\ G \times d_i$;
lbs solute into system = $60\ G \times d_i\ [1-\eta/100]$;
lbs solute out of system = lbs solute into system = $60\ G \times d_i\ \eta/100$
lbs of water exiting the system = $60\ G \times d_i\ \eta/\phi[1 - \phi/100]$
lbs of water evaporated = lbs of water in - lbs of water out = $60\ G \times d_i\ [1 - \eta/100] - (\eta/\phi)[1 - \phi] = 60\ G \times d_i\ [1 - \eta/100]$;

lbs steam required = $60\ G \times d_i / S_1\ [1 - \eta/100]$
where S_1 is defined as the steam economy prior to upgrading equipment. The pounds of steam required after making changes in equipment is:
lbs of steam = $(60\ G \times d_i)/S_2\ [1 - \eta/100]$;
Hence, the change in steam requirement is as follows:
$\Delta S = 60\ G \times d_i\ [1 - \eta/100]\ [1/S_1 - 1/S_2]$
In the above expressions, the parameters are defined as:
η = weight percent solute in the dilute solution
ϕ = weight percent solute in the concentrated solution
S1 = initial steam economy (i.e., lb water evaporated per lb of steam)
S2 = final steam economy (i.e., lb water evaporated per lb of steam).

TEMPERATURE AND LATENT HEAT ESTIMATION FOR SATURATED STEAM

The following expressions can be used to estimate the temperature and enthalpy of steam. The expressions are based upon multiple regression analysis. The equation for temperature is accurate to within ± 1.5 % at 1,000 psia. The expression for latent heat is accurate to within ± 3 % at 1,000 psia. Input data required to use these equations is the steam pressure in psia. The parameters in the equations are defined as t for temperature in °F, H_v for latent heat in Btu/lb, and P for pressure in psia.

$$t = 1.8 \exp\{4.61 + 0.250336\ (\ln(P) - 2.686)\} + 32$$

$$H_v = 1.8\ \{\exp(6.29517 - 0.0343248\ (\ln(P) - 2.686)) - 0.116672P\}$$

ESTIMATING CONDENSATE FOR FLASH TANKS

Often in plant operations condensate at high pressures are let down to lower pressures. In such situations some low-pressure "flash" steam is produced, and the low-pressure condensate is either sent to a power plant or is cascaded to a lower pressure level. The following analysis solves the mass and heat balances that describe such a system, and can be used as an approximate calculation procedure. Refer to Figure 2 for a simplified view of the system and the basis for developing the mass and energy balances. We consider the condensate to be at pressure P_1 and temperature t_1, from whence it is let down to pressure P_2. The saturation temperature at pressure P_2 is t_2. The vapor flow is defined as V lbs/hr, and the condensate quality is defined as L lbs/hr. The mass balance derived from Figure 2 is:

$$F = V + L$$

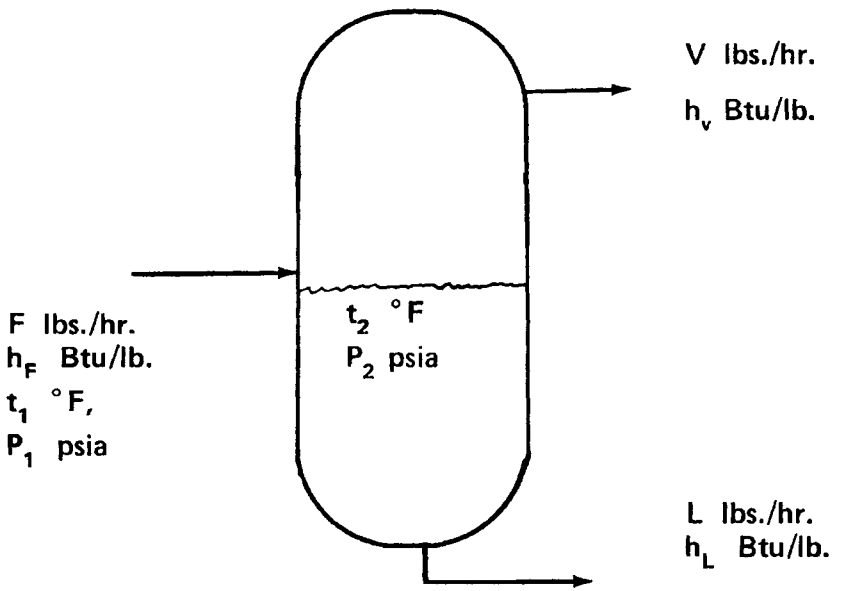

Figure 2. *Material and energy balance diagram for flash tank.*

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And the heat balance is:

$$F \times h_F = V \times h_v + L \times h_L$$

where we define $h_F = t_1 - 32$, and $h_L = t_2 - 32$. The expression may be restated as:

$$F(t_1 - 32) = Vh_v + L(t_2 - 32)$$

Combining the above expressions, we obtain:

$$L/F = \{h_v - t_1 + 32\}/\{h_v - t_2 + 32\}$$

This equation expresses the fraction of the high-pressure condensate returned as low-pressure condensate. The input information needed to apply the working equation are t_2 and P_1. Values for h_v can be approximated from the following:

$$h_v \approx \Delta h_v + h_L = \Delta h_v + t_2 - 32$$

Combining this approximation into the working expression gives:

$$L/F = (\Delta h_v + t_2 - t_1)/\Delta h_v$$

To summarize, the symbols used in the development of the expression are defined as follows:

F = Flow rate of high-pressure condensate, lbs/hr
h_v = Enthalpy of the vapor, Btu/lb
h_F = Enthalpy of high-pressure condensate, Btu/lb
h_L = Enthalpy of low-pressure condensate, Btu/lb
L = Flowrate of low pressure condensate, lb/hr
P1 = Upstream pressure, psia
P2 = Pressure downstream of valve, psia
t_1 = Temperature upstream of letdown valve, °F
t_2 = Temperature downstream of letdown valve, °F
Δh_v = latent heat of vaporization at temperature t_2, Btu/lb.

LINEAR VELOCITY OF AIR THROUGH DUCTS

An estimate of the linear velocity of air through ductwork is a common exercise for process engineers. The calculation requires information on the static pressure drop and volumetric flowrate at any given temperature and pressure.

A simplified estimate can be made by first converting the flow at actual conditions to the flow at standard conditions (i.e., at 70 °F and 1 atm). The calculation basis for the linear velocity assumes a roughness coefficient of 0.0005 and a kinematic viscosity for air of 1.62×10^{-4} ft^2/sec. From the ideal gas law, the following expression is developed:

$$scfm = (acfm)\ (P/T)\ (529)$$

where: scfm = flow, ft^3/min at 70 °F and 1 atm
acfm = flow, ft^3/min at T °R and 1 atm
P = actual pressure, atm
T = absolute temperature, °R

From the standard cubic feet per minute estimate, the linear velocity is as follows:

$$V = 466\ (h^{0.4})\ (scfm)^{0.24}$$

Where V is the linear velocity in fpm and h is the static pressure drop in units of inches of water per 100 ft of pipe.

THERMAL CONDUCTIVITIES OF GASES

Thermal conductivities for gaseous compounds are important in unit operations involving heat transfer coefficients. Thermal conductivities can be readily computed from an empirical polynomial expression that has the following form:

$$K_G = A + BT + CT^2 + DT^3$$

where K_G = Thermal conductivity of the gas at low pressure, (approximately 1 atm). A, B, C, and D are correlation constants that are specific to the chemical, and T is temperature, °K. Correlation constants for different chemical compounds are given in Table 1.

Table 1 provides states the temperature range over which the correlation constants are reported from the literature. Estimates from this expression using the constants in Table 1 are generally accurate and typically provide agreement to well within ± 1% compared to experimentally measured thermal conductivities.

More extensive data can be found in standard references such as Perry's *Chemical Engineering Handbook* or the *CRC Handbook of Chemistry and Physics*. The following table, however will provide some basic information for many common calculations that the process engineer can easily run on his or her PC.

Table 1. *Correlation Constants for Estimating Thermal Conductivity of Gases*

Chemical	A	B x 10^2	C x 10^4	D x 10^8	Range, °C
Halogens					
Flourine, F_2	1.8654	19.79	1.24	-17.77	-30 to 525
Chlorine, Cl_2	3.25	5.8	0.21	-1.25	-80 to 1,200
Bromine, Br_2	-0.16	4.13	-0.03	-0.09	-80 to 1,200
Iodine, I_2	0.63	2.73	-0.03	0.15	-80 to 1,200
Sulfur oxides					
Sulfur dioxide, SO_2	-19.31	15.15	-0.33	0.55	0 to 1,400
Sulfur trioxide, SO_3	-15.96	16.9	-0.47	3.00	-100 to 1,000
Nitrogen oxides					
Nitrous oxide, N_2O	-18.71	21.26	-0.2142	-0.6371	-100 to 1,400
Nitric oxide, NO	11.99	17.18	-0.02	-0.85	-190 to 1,400
Nitrogen dioxide, NO_2	-33.52	26.46	-0.755	1.071	25 to 1,400
Carbon oxides					
Carbon monoxide, CO	1.21	21.79	-0.8416	1.958	-160 to 1,400

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Table 1 Continued.

Chemical	A	B x 10^2	C x 10^4	D x 10^8	Range, °C
Carbon dioxide, CO_2	-17.23	19.14	0.1308	-2.514	-90 to 1,400
Hydrogen halides					
Hydrogen fluoride, HF	9.21	12.60	0.54	-2.34	-100 to 1,400
Hydrogen chloride, HCl	-0.26	12.67	-0.25	0.16	-150 to 1,400
Hydrogen bromide, HBr	-1.89	9.16	-0.26	0.53	-150 to 1,400
Hydrogen iodide, HI	-5.14	7.28	-0.22	0.43	-150 to 1,400
Nitrogen hydrides					
Ammonia, NH_3	0.91	12.87	2.93	-8.68	0 to 1,400
Hydrazine, N_2H_4	-53.89	28.50	0.20	-0.19	0 to 1,400
Hydrogen oxides					
Water, H_2O	17.53	-2.42	4.3	-21.73	0 to 800
Hydrogen peroxide, H_2O_2	-21.07	16.97	0.17	-1.56	0 to 1,200
Diatomic gases					
Hydrogen, H_2	19.34	159.74	-9.93	37.29	-160 to 1,200
Nitrogen, N_2	0.9359	23.44	-1.21	3.591	-160 to 1,200
Oxygen, O_2	-0.7816	23.8	-0.8939	2.324	-160 to 1,200
Inert gases					
Helium, He	88.89	93.04	-1.79	3.09	-160 to 800
Neon, Ne	21.75	36.81	-2.005	6.042	-160 to 1,200
Argon, Ar	6.48	13.23	-0.52	1.32	-160 to 1,200
Olefins					
Ethylene, C_2H_4	-42.04	28.65	0.7963	-3.262	-75 to 1,000
Propylene, C_3H_6	-18.11	14.57	2.38	-9.17	-100 to 1,000
1-Butene, C_4H_8	-25.12	13.78	2.43	-10.20	-100 to 1,000

Table 1 Continued.

Chemical	A	B x 10^2	C x 10^4	D x 10^8	Range, °C
		Alkanes			
Methane, CH_4	-4.463	20.84	2.815	-8.631	0 to 1,000
Ethane, C_2H_6	-75.8	52.57	-4.593	39.74	0 to 750
Propane, C_3H_8	4.438	-1.122	5.198	-20.08	0 to 1,000
		Xylenes			
o-Xylene, $C_6H_4(CH_3)_2$	-13.66	8.53	1.78	-6.26	0 to 1,000
m-Xylene, $C_6H_4(CH_3)_2$	31.53	-10.02	3.97	-14.58	0 to 1,000
p-Xylene, $C_6H_4(CH_3)_2$	-19.53	9.29	1.81	-6.93	0 to 1,000
		Aromatics			
Benzene, C_6H_6	-20.19	8.64	2.34	-9.69	0 to 1,000
Naphthalene, $C_{10}H_8$	-22.40	11.79	0.91	-2.54	0 to 1,000
		Benzene derivatives			
Aniline, $C_6H_5NH_2$	-26.39	11.89	1.55	-4.30	0 to 1,000
Phenol, C_6H_5OH	-31.87	15.26	1.74	-4.40	0 to 1,000
Chlorobenzene, C_6H_5Cl	-15.27	6.29	1.75	-5.53	0 to 1,000
		Alkyl aromatics			
Toluene, $C^6H_5CH_3$	18.14	-9.57	5.66	-22.22	0 to 1,000
Ethylbenzene, $C_6H_5C_2H_5$	1.44	-1.40	5.11	-21.31	0 to 1,000
Cumene, $C_6H_5CH(CH_3)_2$	-13.35	5.38	4.33	-17.92	0 to 1,000
		Cycloalkanes			
Cyclopropane, C_3H_6	-20.46	9.74	3.77	-16.28	0 to 1,000
Cyclobutane, C_4H_8	-23.39	9.15	3.52	-14.81	0 to 800
Cyclohexane, C_6H_{12}	-20.57	4.45	4.07	-17.31	0 to 800
Cyclopentane, C_5H_{10}	-20.35	5.91	3.87	-16.51	0 to 800

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DETERMINING PSEUDOCRITICAL PROPERTIES

Critical properties of gaseous compounds are useful in determining the P-V-T (*Pressure-Volume-Temperature*) properties at nonideal conditions. The compressibility factor Z is defined by the following relationship:

$$PV = nZRT \; PV = nZRT$$

where n is the number of moles and R is the universal gas law constant. To calculate Z, the critical properties must be known. Knowing the critical pressure and temperatures, charts for computing Z can be used.

When the compounds involved are gaseous mixtures, a weighting method is used to arrive at the critical properties of the mixture.

Table 2 illustrates the weighting method that can be used. The required input of parameters to perform computations are the number of components, the mole fraction of ith component, the critical pressure of ith component in units of atm, the critical temperature of the ith component (°K), the molecular weight of the ith component, the pressure of the mixture (atm), the temperature of the mixture (°F), and the compressibility factor of the mixture, Z_m.

The example provided in Table 2 is for a four component mixture of hydrocarbons (methane, ethane, propane, and n-octane). The weighting method is a common calculation procedure that process engineers will encounter many times. Computations for simple systems can be easily set up on an Excel spreadsheet.

Table 2. *Weighting Method for Estimating Critical Properties of Organics*

Compound	Mole Fraction	Critical Press. (atm)	Critical Temp. (°R)	Pseudocritical Press. [(a)]	Pseudocritical Temp.[(b)]
Methane	0.20	45.4	190.6	9.080	38.12
Ethane	0.30	48.2	305.4	14.460	91.62
Propane	0.35	41.9	369.8	14.670	129.43
n-Octane	0.15	24.5	568.8	3.675	85.32
				41.885	344.49
P_c (mixture) = 41.885 atm.					
T_c (mixture) = 344.490 °K					

(a) ...Col. 2 x Col. 3; (b) ...Col. 2 x Col. 4.

The following formulas can be used to calculate critical properties:

$$T_{cm} = \sum T_{ci} y_i$$
$$P_{cm} = \sum Pc_i y_i$$
$$M_m = \sum M_i y_i$$
$$T_{rm} = T_m / T_{cm}$$
$$P_{rm} = P_m / P_{cm}$$
$$V_r = Z_m R T_{cm} / P_{cm} M_m$$

In the above formulas the terms are defined as follows:

T_m = Temperature of the mixture, °R
T_{cm} = Critical temperature of the mixture, °R
T_{ci} = Critical temperature of the ith component, °R
P_m = Pressure of the mixture, atm
P_{cm} = Critical pressure of the mixture, atm
P_{ci} = Critical pressure of mixture, atm
M_m = Molecular weight of mixture
M_i = Molecular weight of ith component
y_i = Mole fraction of ith component
R = Universal gas law constant = 0.73 (atm)(ft^3)/(lb-mol)(°R)
T_{rm} = Reduced temperature of mixture
P_{rm} = Reduced pressure of mixture
Z_m = Compressibility factor of mixture

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ESTIMATING HEAT EXCHANGER TEMPERATURES

Chapter 1 provides a summary of important equations for estimating the terminal temperatures in a heat exchanger. Here we formalize a short estimating procedure for a countercurrent flow situation. Assume that a specifier of a heat exchanger has defined a preliminary sizing of the unit. The system requires heat and material balances.

The only information available are the inlet temperatures of the hot and cold fluids and estimates for the overall heat transfer coefficient U and the heat transfer surface A. The flows are fixed and the specific heats of the fluids known.

The problem is different from typical heat transfer problems. The heat balance is not straightforward because the outlet temperatures are unknown. A trial-and-error procedure is therefore required.

Before this procedure is set up, the convergence equations have to be established as follows (Refer to Figure 3):

$$W_1 C_1 (T_1 - t_1) = W_2 C_2 (T_2 - t_2) = UA \Delta T_{LMTD}$$

where W_1 and W_2 are mass flows of the cold and hot fluid in lbs/hr, respectively.

Parameters C_1 and C_2 are specific heats of the cold and hot fluids in Btu respectively, and the log-mean temperature difference is defined as follows:

$$\Delta T_{LMTD} = \{(T_2 - T_1) - (t_2 - t_1)\}/\{\ln[(T_2 - T_1)/(t_2 - t_1)]\}$$

Rearranging this expression gives the following:

$$(W_1C_1/W_2C_2)(T_1 - t_1) = T_2 - t_2$$

$$T_2 = T_2 - aT_1 + at_1 = (T_2 + at_1) - aT_1$$

where $a = W_1C_1/W_2C_2$. Defining the following parameter and substituting into the above expression, $b = T_2 + at_1$:

$$\Delta T_{LMTD} = \{(T_2 - T_1) - (b - aT_1 - t_1)\}/\ln[(T_2 - T_1)/(b - aT_1 - t_1)]$$

The heat transfer rate can be determined from the following equation:

$$Q = UA\ [(T_2 - T_1) - (b - aT_1 - t_1)]\ /\ [\ln\{(T_2 - T_1)\ /\ (b - aT_1 - t_1)\}]$$

The above working expression may also be stated in the following form:

$$W_1C_1\ (T_1 - t_1) = UA\ [(T_2 - T_1) - (b - aT_1 - t_1)]\ /\ [\ln\{(T_2 - T_1)\ /\ (b - aT_1 - t_1)\}]$$

And upon simplifying further, we obtain:

$$W_1C_1\ (T_1 - t_1)\ [\ln(T_2 - T_1)\ /\ (b - aT_1 - t_1] = UA\ (T_2 - T_1) - (b - aT_1 - t_1)$$

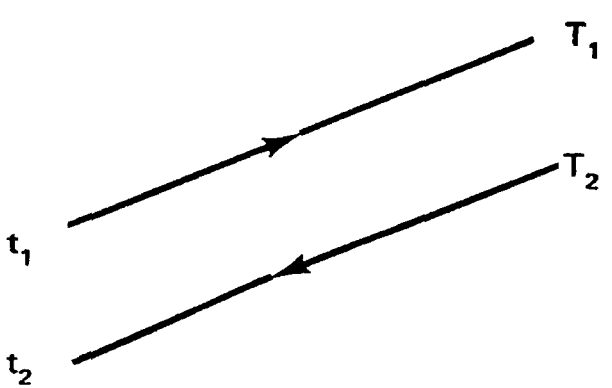

Figure 3. *Terminal temperatures in a counterflow heat exchanger.*

The last expression represents the working formula, whereby the calculation method assumes a value of T_1, calculating both sides of the above expression until they converge.

ESTIMATING THE VISCOSITY OF GASES

The following polynomial can be used to estimate the viscosity of gases:

$$\mu = A + BT + CT^2$$

where: μ = Gas viscosity, micropoise, and A, B, and C are constants. The calculation procedure used here is similar to that explained in the section on estimating thermal conductivity. The only change is the degree of the polynomial, which is now two. Table 3 provides reference viscosity data for gases at 25 °C, and Table 4 provides correlation constants for various common gases encountered throughout the CPI. The viscosities of mixtures can be computed from the following relation:

$$\mu_m = \sum y_i \mu_i (M_i)^{1/2} / \sum y_i (M_i)^{1/2}$$

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where: μ_m = Viscosity of mixture
μ_i = Viscosity of the ith component
y_i = Mole fraction of the ith component
M_i = Molecular weight of the ith component

The summations are done from the ith to N number of components. The reader will encounter many situations where it is necessary to estimate the viscosities of mixtures. This is especially the case when calculations involve heat transfer coefficients which requires knowledge of the viscosity of fluids involved. Unlike estimates for densities, viscosity functions tend to be non-linear. The formula provided for estimating the viscosity of a mixture is only reliable for low pressures, and should not be applied to pressures above 100 atm. The term viscosity is a fundamental physical property that has great importance in fluid dynamics as well as heat transfer. Its definition stems from the property of any fluid. A fluid is a substance which undergoes continuous deformation when subjected to a shear stress. The resistance offered by a real fluid to such deformation is called its consistency. If τ (lb. force/ft^2) is the shear stress which maintains at a point in a fluid the time rate of shearing strain du/dy (i.e., velocity gradient, (ft/sec)/ft perpendicular to the direction of τ), the numerical measure of the consistency is the quantity τg_c/ du/dy [lb-mass / (sec)(ft)] in which g_c is the dimensional constant 32.1740 (lb-mass)(ft)/(lb-force)(sec^2). For gases and for Newtonian liquids the consistency is constant if the static pressure and temperature are constant or fixed;

for such substances the consistency is usually called the viscosity (= absolute viscosity). If two parallel plates are separated by a thin layer, L ft. thick, of a fluid of viscosity μ lb-mass/(sec)(ft), the force τ per unit area (lb-force/ft^2) which must be exerted in order to keep one plate moving, parallel to the other at a relative velocity of 1 ft/sec is $\tau = \mu/g_cL$. If the consistency of a fluid is not constant but is, at constant temperature, a function only of the shear stress τ, the fluid is called a complex (or non-Newtonian) fluid. An ideal or perfect fluid is a hypothetical liquid or gas which offers no resistance to shear and therefore has zero consistency. Such an imaginary perfect fluid is not to be confused with a "perfect gas," which is a fluid for which PV = RT, where P is the absolute static pressure, V is the specific volume, R is the gas constant, and T is the absolute temperature. In many cases highly incorrect results are obtained if the viscosity is neglected, although in the same problem the relation PV-RT may be safely used. The unit of viscosity (i.e., absolute viscosity) is the poise (poise = 1 (dyne)(sec)/cm^2 = 1 g/(sec)(cm)). Viscosity data are normally tabulated in centipoises (cp) or micropoises as in Table 3. In the absolute system of units, the dimensionless constant g_c is simply unity.

Table 3. *Viscosity (micropoise) of Common Gases at 25 °C*

Compound	μ_G	Compound	μ_G	Compound	μ_G
Flourine	232.6	Hydrogen iodide	178.2	1-Butene	74.3
Chlorine	133.5	Ammonia	103	Methane	111.9
Bromine	153.8	Hydrazine	80.13	Ethane	92.2
Iodine	136.5	Water	90.14	Propane	82.4
Sulfur dioxide	128.2	Hydrogen peroxide	95.2	o-Xylene	64.8
Sulfur trioxide	138.6	Hydrogen	84.7	m-Xylene	56.7
Nitrous oxide	158.2	Nitrogen	169.5	p-Xylene	58.4
Nitric oxide	192.8	Oxygen	199.2	Benzene	74.6
Carbon monoxide	165.2	Helium	195.7	Naphthalene	53.2
Carbon dioxide	153.4	Neon	306.5	Toluene	68.8
Hydrogen flouride	110.8	Argon	223.3	Ethylbenzene	63.2
Hydrogen chloride	114.2	Ethylene	101.2	Cumene	58.8
Hydrogen bromide	184.5	Propylene	83.9	Phenol	79.0

Table 4. *Correlation Constants for Estimating Gas Viscosity*

Compound	A	B x 10^2	C x 10^6	Appl.Temp. Range,°C
Flourine	22.09	76.9	-211.6	-200 to 1,000
Chlorine	5.175	45.69	-88.54	-200 to 1,200
Bromine	2.153	54.50	-122.2	-200 to 1,200
Iodine	-17.75	54.71	-99.70	-200 to 1,200
Sulfur dioxide	-3.793	46.45	-72.76	-100 to 1,400
Sulfur trioxide	4.207	47.12	-68.34	-100 to 1,400
Nitrous oxide	32.28	44.54	-77.08	-150 to 1,400
Nitric oxide	56.77	48.14	-84.34	-150 to 1,400
Carbon monoxide	32.28	47.47	-96.48	-200 to 1,400
Carbon dioxide	24.45	45.49	-86.49	-100 to 1,400
Hydrogen fluoride	-19.21	45.98	-79.96	-120 to 1,400
Hydrogen chloride	-9.554	54.45	-96.56	-120 to 1,400
Hydrogen bromide	-23.37	74.03	-144.8	-120 to 1,200
Hydrogen iodide	-17.65	69.77	-136.5	-120 to 1,400
Ammonia	-9.372	38.99	-44.05	-200 to 1,200
Hydrazine	-17.05	34.01	-47.51	-200 to 1,400
Water	-31.89	41.45	8,272	0 to 1,000
Hydrogen peroxide	5.381	28.98	38.40	-80 to 1,000
Hydrogen	21.87	22.2	-37.51	-160 to 1,200
Nitrogen	30.43	49.89	-109.3	-160 to 1,200
Oxygen	18.11	66.32	-187.9	-160 to 1,000
Helium	54.16	50.14	-89.47	-160 to 1,200
Neon	87.79	78.60	-176.2	-160 to 1,200
Argon	43.48	63.99	-128	0 to 1,200
Ethylene	3.586	35.13	-80.55	-100 to 800

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Note that the kinematic viscosity of a fluid is defined as the ratio of its viscosity to the fluid density. The c.g.s. unit of kinematic viscosity is usually called the stoke, and is equal to 1 cm^2/sec.

ESTIMATE FOR MECHANICAL DESUPERHEATERS

The chemical process industry relies on mechanical desuperheaters when there is a need to reduce the temperature of superheated steam. One way of accomplishing this is by injecting feedwater quality water or condensate into the steam header at about 100 psi greater than the header pressure. The design of the superheater is based on material and energy balances. Important parameters to be specified are the steam flows into the unit, and the temperatures of the steam in and out of the system. The material and energy balance diagram for a mechanical desuperheater is shown in Figure 4. The following parameters are defined:

h_1 = Enthalpy of steam in
h_1m_1 = Total Btu in
h_2m_2 = Total Btu out in steam
$h_1m_1 - h_2m_2$ = Difference in Btu

The difference is the heat that is picked up by the injected water, and hence we may write:

$$m_w[C_p(t_1 - t_2) + L_H] = m_1h_1 - m_2h_2$$

From a mass balance we write:

$$m_2 = m_1 + m_w$$

Combining the above expressions and solving for the mass of desuperheating water used, we arrive at the following working expression:

$$m_w = m_1(h_1 - h_2) / [C_p(t_1 - t_2) + L_H + h_2]$$

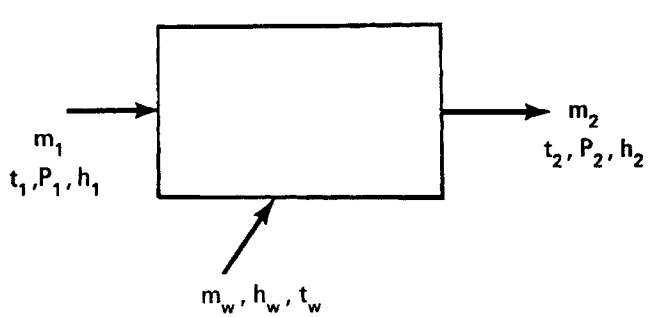

Figure 4. *Mass and energy balance for desuperheater system.*

In the above analysis the parameters are as follows:

C_p = Specific heat of the desuperheating water, Btu/lb-°F
h_1 = Enthalpy of superheated system, Btu/lb
h_2 = Enthalpy of saturated steam, Btu/lb
L_H = latent heat of vaporization of water at t_w, Btu/lb
m_1 = Mass of superheated steam into the system, lb/hr
m_2 = Mass of saturated steam out, lb/hr
m_w = Mass of desuperheating water used, lb/hr
t_1 = Temperature of superheated steam, °F
t_2 = Temperature of saturated steam, °F
t_w = Temperature of desuperheating water, °F

ESTIMATING PUMP HEAD WITH NEGATIVE SUCTION PRESSURE

Figure 5 illustrates the usual configuration for determining the total head developed by a pump working with suction pressure below atmospheric conditions. Suction head is measured by a mercury U-tube which is connected to the suction through an air-filled tube. Discharge head is measured by means of a Bourdon gage connected to the pump discharge through a water-filled tube. The working equations are for the head and the horsepower requirements:

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$$H = 144P_g/\gamma + \gamma_m h_{sg} + Z_d + Z_s + 0.00001056\ (gpm)\ (d_d^2 - d_s^2)$$

$$HP = (gpm) \times (H) \times (\zeta) / 3960$$

where: gpm = Fluid flow rate
ζ = Specific gravity of the liquid
H = Total pump head, ft
P_g = Discharge gage reading, psi
γ = Specific weight of liquid being pumped, lb/ft^3
= (32.2) (specific gravity of liquid)
γ_m = Specific weight of mercury, lb/ft^3
= (32.2) (specific gravity of mercury)
Z_d, Z_s = Elevation at discharge and suction, respectively
gpm = Flow rate of liquid being pumped, gal/min
d_d = Diameter of discharge pipe, in.
d_s = Diameter of suction pipe, in.

The reader should refer to the chapter on pumps for more detailed calculation methods and for a description of common pumping equipment and pumping applications. The simple relationships provided here are useful in obtaining initial design specifications information in sizing a pump for an application.

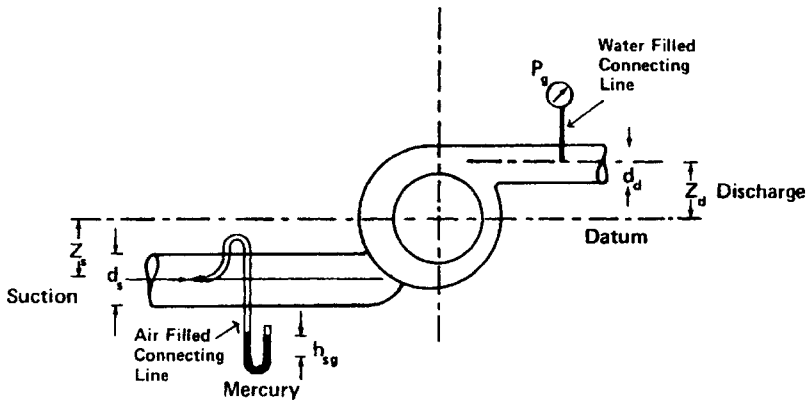

Figure 5. *Diagram for determining head for a pump with negative suction pressure.*

CALCULATIONS FOR BACK-PRESSURE TURBINES

Back-pressure power is achieved by letting down steam at high pressure and temperature for process use. Since process demands change with time and plant conditions, it is of interest to investigate how much power can be obtained at the generator terminals for given quantities of extraction and exhaust steam. Figure 6 illustrates a generalized steam flow configuration for a double-extraction condensing turbine. The outlet enthalpies are measured upstream of any desuperheating stations. The condensing cooling water removes an amount of heat that is equal to (W_3 x h_3) Btu/hr. The working equations for this analysis are:

$$W = W_1 + W_2 + W_3$$

$(3143)/(h_i - h_1)$ = (Theoretical lbs of steam)/(kW generated at 1st extraction point)

$(3143)/(h_i - h_2)$ = (Theoretical lbs of steam)/(kW generated at 2st extraction point)

$(3143)/(h_i - h_3)$ = (Theoretical lbs of steam)/(kW generated at condensing section)

Since each section will have a certain isoentropic efficiency, the actual steam rate for each will be different. The actual steam rates can be expressed by the formulas presented below.

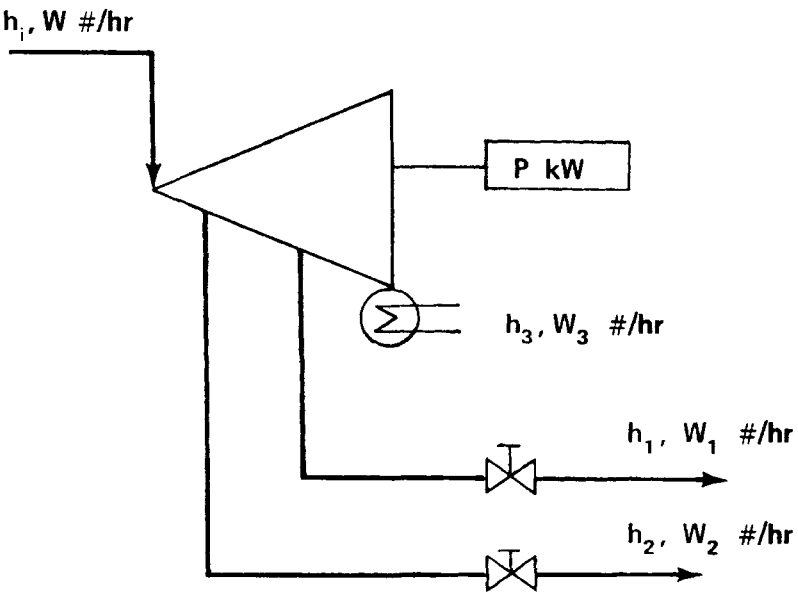

Figure 6. Simplified diagram for back-pressure turbine analysis.

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First extraction point = $(3143) / [(h_i - h_1)e_1]$

Second extraction point = $(3143) / [(h_i - h_2)e_2]$

Condensing stage = $(3143) / [(h_i - h_3)e_3]$

From the above, the total output less generator loss is expressed by the following:

$$(h_i - h_1)e_1W_1/(3143) + e_2W_2(h_1 - h_2)/(3143) + e_3W_3(h_1 - h_3)/(3143) = P/e_g$$

In the above working equation parameter e_g denotes the generator efficiency which usually reaches around 98 %. Note that a first pass estimate can be made by assuming e_1, e_2, and e_3 to remain constant. A more rigorous analysis can be made by varying these efficiencies using data supplied by the equipment supplier. The initial information required for the analysis are the flows, the efficiencies, and the enthalpies. The outputs from the calculations are the total kW at the generator terminals and the overall water rate for the machine This type of analysis is particularly well suited for sensitivity studies of the type used in determining steam-power balances in an operating facility.

TUBESIDE FOULING RATES IN HEAT EXCHANGERS

A well-known empirical formula for fouling in tubes is:

$$R_t = R_o(1 - e^{-At})$$

where: R_t = Fouling resistance at time t, hr-ft^2-°F/Btu
R_o = Asymptotic value of fouling resistance, hr-ft^2-°F/Btu
t = Time corresponding to the fouling resistance R_t
A = Constant describing the rate of fouling, (time)$^{-1}$

A heat exchange surface becomes fouled when the material deposited on it resists the passage of heat and reduces the conductive part of the heat transfer. When this occurs in practice, the exchanger is taken off line and cleaned until it fulfills the heat duty for which it was designed. It is useful to know the frequency at which the cleaning must be performed. If the downtime for the exchanger is minimized, the operating costs for the system are minimized. The following method uses a trial-and-error procedure that predicts fouling rates and resistance over given periods of time. In the above expression R_o and A are unknown. What are known (or can be found experimentally) are the values of R_t at different times. For example, field data on terminal temperatures and flow rates taken on two different days would yield R_1 and R_2 at time t^1 and t_2 where $t_2 > t_1$ and $R_2 > R_1$. This gives us the following relationships:

$$R_1 = R_o (1 - e^{-At1}) \text{ or } R_o = R_1 / (1 - e^{-At1})$$

$$R_2 = R_o (1 - e^{-At2})$$

Combining these two expressions gives:

$$R_2/R_1 (1 - e^{-At1}) = 1 - e^{-At2}$$

We introduce the following:

$$1 - \alpha = e^{-At2} - \alpha e^{-At1},$$

where $\alpha = R_2/R_1$

Whereupon, the final working relationship is obtained:

$$(1 - \alpha) - e^{-At2} + \alpha e^{-At1} = 0$$

The working expression can be solved by trial and error to obtain a value for A. Once parameter A is known, the preceding expression can be used to calculate R_o. The input parameters needed for the analysis are R_1, R_2, t_1, and t_2.

CALCULATIONS FOR PIPE FLOWS

Estimating Pressure Drop for Turbulent Flow

Pressure drop calculations for simple geometries like pipe flows are straightforward. The Fanning friction factor is given by:

$$f = 0.054/Re^{0.2}$$

where: f = Fanning friction factor
Re = Reynolds number

This correlation is good for a Reynolds number range of between 2,100 and 10^6. The Fanning equation for pressure drop is:

$$\Delta P_f = 2fLV^2\rho/g_cD$$

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where: ΔP_f = Frictional pressure drop, lb/ft^2
L = Pipe length, ft
V = Velocity of fluid, ft/sec
D = Pipe internal diameter, ft
ρ = Fluid density, lb/ft^3
gc = Gravitational constant, 32.2 lb_m-ft/lb_f-sec^2

Combining both expressions gives a simple relation for estimating the frictional pressure drop in straight run pipe.

$$\Delta P_f = 2\,[(0.054)/Re^{0.2}]\,[LV^2\rho / g_cD]$$

Determining Economic Pipe Diameter

The following analysis can be used to determine economic pipe diameters for the turbulent flow of Newtonian fluids. The working expression that can be used is:

$$D^{4.84+n} / [1 + 0.794L_eD] =$$
$$\{0.000189\ YKq^{2.8}\rho^{0.84}\mu^{0.16}((1 + M)(1 - \Phi) + ZM/(a' + b'))\}/\{n \times E(1 + F)[Z + (a + b)(1 - \Phi)]\}$$

The above expression can be solved by a trial and error calculation, whereby the following key assumptions are applied: (1) the flow is turbulent, i.e., Re $\geq$ 2,100, and (2) motive power is supplied by a prime mover such as a pump or a compressor. To make this expression easier to use, the terms are rearranged in the following manner:

$$P(D) = 0$$

where: $P = D^m - R - RSD$
$m = 4.84 + n$
$S = 0.794L_e$
R = Right Hand Side of the Working Equation

The symbols in the working equation are defined as follows:

D = Economical pipe diameter, ft
L_e = Factor for friction in fittings, equivalent length in pipe diameter per unit length of pipe (1/ft)
n = Exponent in pipe cost equation, $C = XD^n$; Note that C = Cost of pipe, \$/ft; X = Cost of 1 ft of 1 ft diameter pipe, \$/ft
$M = EP(a' + b')/17.9\ KY$
where: E = Combined fractional efficiency of pump and motor
P = Installed cost of pump and motor, \$/hp
K = Cost of power delivered to the motor, \$/kW-hr
Y = Days of operation per year
a = Fractional annual depreciation on pipe line (dimensionless)
b = Fractional annual maintenance on pipe line (dimensionless)
Z = Fractional annual rate of return on incremental investment
q = Flow rate, ft^3/sec
F = Ratio of cost of fittings plus installation cost of pipe and fittings to cost of pipe only
a' = Fractional annual depreciation on pumping installation
b' = Fractional annual maintenance on pumping installation
ϕ = Factor accounting for taxes
μ = Fluid viscosity, cp
ρ = Fluid density, lb/ft^3

Estimating Gas Temperatures in Pipelines

Gas temperature is an important quantity when designing gas transmission lines, since pipelines are designed for large flows and their consequent higher compressor requirements. The compressor power required at each station depends to a great extent on the gas temperature. Flowing gas temperature must be controlled to a

certain degree, particularly in very cold (Arctic) and very hot (Sahara) regions. The following analysis from the literature enables a estimating method for determining gas temperatures in pipelines. The steady state flow of natural gas in pipelihes obeys the first law of thermodynamics for a compressible fluid; which can be simply stated as follows:

$$m(dv/dx) + g(dz/dx) + (dh/dx) + (dq/dx) + (dw/dx) = 0$$

The enthalpy term h includes both internal energy and flow work terms. Since dw=0 between compressor stations, the equation becomes:

$$m(dv/dx) + g(dz/dx) + (dh/dx) = (dq/dx)$$

Neglecting changes in kinetic energy and potential energy produces:

$$dh/dx = dq/dx$$

For a compressible fluid that undergoes exansion through a valve or an orifice, the Joule-Thompson coefficient is defined as:

$$\mu = (\delta T/\delta P)_h$$

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The relationship between heat content, temperature and pressure is defined by the following:

$$\begin{aligned} dh &= (\delta T/\delta P)_p dT + (\delta T/\delta P)_T dP \\ &= (\delta T/\delta P)_p dT + (\delta T/\delta P)_h (\delta T/\delta P)_p dP \\ &= (\delta T/\delta P)_p dT + \mu C_p dP \\ &= C_p dT + \mu C_p dP \end{aligned}$$

The conductive heat flow to the pipeline from the surrounding solid is:

$$dq/dx = (2\Pi UR/m)(T_g - T)$$

And combining the above expressions we obtain:

$$C_p (dT/dx) + \mu C_p (dP/dx) = (2\Pi UR/m)(T_g - T)$$

Defining $n = (2\Pi UR)/(mC_p)$, and introducing n into the above expression, we obtain an expression that after some simplification and rearrangement of terms that is based on assuming n and dP/dx to be constants:

$$dT/dx = -n\,(T - T_g) - \mu(dP/dx) = -n[T - T_g + (\mu/n)(dP/dx)]$$

This expression can be solved to provide:

$$\ln[T - T_g + (\mu/n)(dP/dx)] = -nx + C_1$$

The following boundary condition is applied: $x = 0$, $T = T_1$; and the solution is simplified to the following:

$$T = (T_1 - T_g + b)e^{-nx} + T_g - b$$

This last expression represents the working equation for the analysis, in which the terms are defined as follows:

b = Parameter defined as $(\mu/n)(dP/Dx)$
C_p = Specific heat of gas, Btu/lb-°F
g = Gravitational constant, 4.17×10^8 $(lb_m/lb_f)(ft/hr^2)$
h = Molal flow rate of gas, lb/hr
m = Mass flow rate of gas, lb/hr
n = Parameter defined as $2\delta UR/m\,C_p$
P = Pressure of gas at any section, psi
q = Heat flow to gas by conduction, convection and radiation, Btu/hr-lb
R = Radius of pipeline, ft
T_x = Temperature of the flowing gas at any section x-ft from the reference point, °F
T_1 = Temperature of the gas at the point of origin, °F
W = Thermodynamic flow work term
U = Heat transfer coefficient for transfer between the pipeline and surrounding ground, Btu/hr-ft^2-°F
X = Distance from origin, ft
Z = Distance above datum, ft
μ = Joule-Thompson coefficient
δ = Incremental amount of a variable

Pipeline Diameter Calculation for Compressible Gas Flows

The following analysis enables one to calculate the diameter of a pipeline transporting any compressible fluid. The required inputs are volumetric flow rate, the specific gravity of the gas relative to air, flow conditions, compressibility factor Z where Z is defined by $nZRT = PV$, the pressure at the point of origin and the destination, the pipe length, and pipe constants such as effective roughness. The working equations have been obtained from the literature. Since the friction factor

is an exponential function of the flow and therefore, the diameter, equations relating diameter, pressure drop, and flow are complex. The calculation procedure involves a trial-and-error method to compute the internal diameter, with the working equation being:

$$Q_b = 38.77\ (T_b/P_b)\ [(1/G)\ (1/T_f)\ (1/Z)]^{1/2} D^{2.5} [(P_1^2 - P_2^2)/L]^{1/2}\ (1/f)^{1/2}$$

where: Q_b = Flow, scfd at base conditions
T_b = Base temperature, 520 °R
P_b = Pressure base, 14.73 psia
G = Specific gravity of gas, relative to that of air = 1
T_f = Flowing temperature, average °R
Z = Supercompressibility, average for pipeline segment or at
$P_{avg} = 2/3\ [P_1 + P_2 - P_1P_2/(P_1 + P_2)]$
P_1 = Inlet pressure, psia
P_2 = Outlet pressure, psia
D = Inside pipe diameter, inches
L = Segment length, miles
f = Friction factor, dimensionless

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Friction factor f is defined by $(1/f)^{1/2} = 4 \log (3.7\ D/K_e)$, and K_e is equivalent pipe roughness, inches. The working equation may be simplified to the following:

$$Q_b = MD^{2.5} \log (3.7\ D/K_e)$$

where: $M = 5476\ (G\ T_f\ Z)^{-1/2}\ [(P_1^2 - P_2^2)/L]^{1/2}$

In performing computations, values should be tabulated as quantities of Q_b/M and $D^{2.5} \log(3.7D/K_e)$. If the difference in the two values is large, then a new guess of the diameter D is needed, or the value of the iteration increment must be changed.

Estimating Friction Factors

Determination of friction factors for some fluid flow applications can involves a trial-and-error procedure because the friction factor is not a simple function of the Reynolds number. Process engineers, therefore, refer to a Moody chart that has been developed using the following relationships:

$$f = 64/Re, \text{ for } 1 \leq Re \leq 2100, \text{ and}$$
$$1/\sqrt{f} = -2 \log\ [2.51/Re\sqrt{f} + \epsilon/3.7D], \text{ for } 2100 \leq Re \leq 10^6$$

In theses equations f is the friction factor, Re the Reynolds number, and ϵ/D is the relative roughness of the conduit. Inspection of the second equation reveals that for

turbulent flow, calculation of the friction factor involves a trial-and-error procedure. To avoid this method, another expression may be used to calculate the friction factor in all three flow regimes: laminar, transition, and turbulent. This expression is accurate for most engineering applications. The full-range friction factor is given as:

$$f = [(8/Re)^{1/2} + 1/(A + B)^{1.5}]^{1/12}$$

where parameters A and B are defined as follows:

$$A = [2.457 \ln\{1/\{(7/Re)^{0.9} + 0.27\,\epsilon/D\}\}]^{16}$$

$$B = [37{,}530/Re]^{16}$$

Hence, to determine the friction factor, the input information needed is the relative roughness ϵ/D and the Reynolds number.

Petroleum Liquid Flows in Pipelines

The flow of petroleum liquids may be analyzed the same way as any other fluid flow problem. However, due to some industry convention, it is necessary to formulate an exclusive set of working equations. Examples of the "traditional ways" are expressing specific gravity in API units, viscosity in centistokes, flow in barrels per day, etc. Regular algorithms for fluid flow problems could very well be used, but the process engineer would have to convert the units before the making calculations. For this reason, the following estimation method has been designed to accept inputs in units commonly used in the petroleum industry. The working equations have been taken from the available.

$$H/L = C_1Q^2/F^2D^5$$

where: H = Total head loss, ft or meters
L = Total pipe length, miles or km
Q = Volumetric flow rate, bbl/hr or m^3/hr
D = Internal pipe diameter, inches or meters
F = Fanning friction factor
C_1 = Constant = 322.04, U.S. Customary System (USCS)
= 25.508 x 10^{-6}, International System (SI) units

The value of C3 is 0.011454 in USCS units and 20.178 x 10^{-6} in S1 units. The inputs for the calculation are Q (bbl/hr or m^3/hr) and pipeline length (miles or km), viscosity U (Centistokes), pipe diameter D (inches or meters), effective pipe roughness e, and pipeline lengths (miles or km). The Fanning friction factor is

related to the Darcy friction factor by definition:

$$F^2 = 4/f_D$$

Note that the Reynolds number is defined as follows:

$$Re = C2Q/D\mu$$

Where μ is defined as the kinematic viscosity (centistokes), and C_2 is a constant with a value of 2,213.8 in USCS units and 353.68 in SI units. An empirical relation for the Fanning friction factor is the Colebrook-White equation:

$$F = -4 \log (e/3.7D + 1.413 F/R)$$

By combining the above working expressions we obtain the final working expression for the analysis:

$$H/L = (C_1Q^2/D^5) [-4 \log \{(e/3.7D) + (C_3\mu/D^{3/2})(1/(H/L))^{1/2}\}]^{-2}$$

And the constant C_3 is defined as follows:

$$C3 = 1.413 C_1^{1/2} / C_2$$

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RECOVERY IN MULTICOMPONENT DISTILLATION

The following is a simplified estimating procedure for recovery in multicompnent distillation. In the working expressions provided below, the parameters b, d, and f rpresent the bottoms, distillate, and feed, respectively. Subscripts i, HK, and LK represent the component i, the heavy-key component, and the light-key component, repsectively. Relative volatility is represented by symbol α. Calculations can be readily set up on an Excel Spreadsheet.

$$d_i/f_i = 10^A \alpha_i^B / (1 + 10^A \alpha_i^B)$$

$$b_i/f_i = 1 / (1 + 10^A \alpha_i^B)$$

$$A = -\log [(b_{HK}/f_{HK})/(1 - b_{HK}/f_{HK})]$$

$$B = \log [\{(d_{LK}/f_{LK})/(1 - d_{LK}/f_{LK})\}\{(b_{HK}/f_{HK})/(1 - b_{HK}/f_{HK})\}] / \log (\alpha_{LK})$$

Physical properties data can be obtained from standard references such as the *CRC Handbook of Chemistry and Physics*.

ESTIMATING EQUILIBRIUM CURVES

In distillation work for binary systems with constant relative volatility, the equilibrium between phases for a given component can be expressed by the following equation:

$$y = \alpha x / \{1 - (1 - \alpha)x\}$$

where: x = Mole fraction in liquid phase
y = Mole fraction in vapor phase
α = Relative volatility

The procedure to calculate the points for an equilibrium curve is to start with $x = 0$, and calculate the value of y for a given value of α. A second point is calculated by incrementing the value of x and repeating the calculation for y. This calculation is repeated until the value of $x = 1$.

ESTIMATING EVAPORATION LOSSES FROM LIQUIFIED GASES

Liquified gases are sometimes stored in well-insulated spherical containers that are vented to the atmosphere. Examples in the industry are the storage of liquid oxygen and liquid ammonia in spheres. If the radii of the inner and outer walls are r_o and r_i, and the temperatures at these sections are T_o and T_1, an expression for the steady-state heat loss from the walls of the container may be obtained. A key assumption is that the thermal conductivity of the insulation varies linearly with the temperature according to the relation:

$$K = K_o + (K_1 - K_o)\,[(T - T_o) / (T_1 - T_o)]$$

The expression for the steady-state heat loss through the walls of the vessel is:

$$Q = 4\pi ror1[(Ko + K1)/2][(T1 - To)/(r1 - ro)]$$

The inputs required for the calculation are the radii, inner and outer temperatures, and thermal conductivities at the two temperatures. This expression enables an estimate of the heat flow into a spherical storage tank containing liquified gas.

COMBUSTION AIR CALCULATIONS

The following provides a calculation method for determining the amount of air needed for perfect combustion of one cubic foot of any gaseous fuel. The following expression provides an estimate of the ratio of the volume of air needed to the volume of fuel (i.e., the air to fuel ratio, Θ):

$\Theta = [(\%CH_4 \times 0.0956) + (\%C2H_6 \times 0.1673) + (\%C_3H_8 \times 0.239) + (\%C_4H_{10} \times 0.311) + (\%H_2 \times 0.0239) + (\%CO \times 0.0239) - \%O_2 \times 0.0478] \times [1 + XSA/100]$

In this equation all percentages are on a volume basis. The term XSA refers to the excess air over the stoichiometric requirement. The volumes of the air and gas must be measured at the same temperature and pressure. For consistency, it is best to first convert to actual conditions (i.e., from actual cubic feet (ACF) to standard cubic feet (SCF). The following formula can be used for this conversion:

$$SCF = (ACF)\ (36.05)\ (P_1 + 14.7) / (t_1 + 460)$$

The ACF is the actual cubic feet of gas measured at t_1 °F and P_1 psig. SCF represents standard conditions at 70 °F and 14.6 psia. The formulas provided require input information on the pressure and temperature of the fuel gas, the fuel gas analysis by volume (or mole percent if the pressures are sufficiently low), and the percent excess air. The calculation provides the air to fuel ratio required for complete combustion.

ESTIMATING TEMPERATURE PROFILES IN AGITATED TANKS

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A common process task involves heating a slurry by pumping it through a well-stirred tank. It is useful to know the temperature profile of the slurry in the agitated vessel. This information can be used to optimize the heat transfer process by performing simple sensitivity studies with the formulas presented below. Defining the inlet temperature of the slurry as T_i, and the temperature of the outer surface of the steam coil as T_s; then by a macroscopic mass and energy balance for the system, a simplified calculation method is developed.

The system configuration is shown in Figure 7. The following system parameters are defined:

V = Volume of slurry in tank, ft^3

ρ = Density of slurry, lb_m/ft^3

W = Mass flow rate of slurry through tank, lb/hr

C_P = Heat capacity of slurry, $Btu/(lb_m)$ (°F)

U = Overall heat-transfer coefficient of heating coil Btu (ft^2) (°F) (hr)

A = Total heat-transfer surface of coil, ft^2

t = Time since start of heating

T(t) = Temperature of slurry in tank

Perfect mixing is assumed. The temperature profile of the tank with respect to time is then given by the following expression reported in the literature.

$$(T-T_\infty) / (T_i - T_\infty) = \exp[-\{(UA/\rho VC_p) + W/\rho V\}t]$$

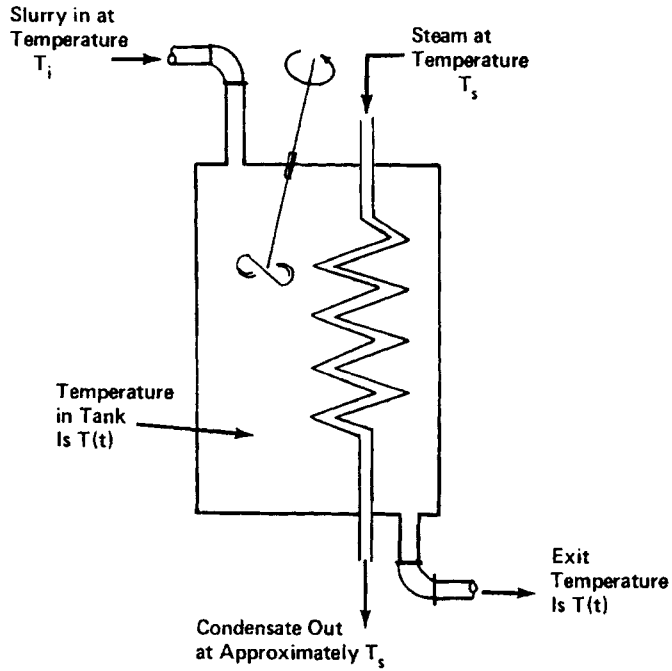

Figure 7. *System diagram for heated mixing tank.*

where:

$$T_\infty = [(UA/WC_p)T_s + T_i]/[1 + (UA/WC_p)]$$

By computing values from the above equations, a curve of T versus time may be plotted. This information may be used to design heating systems or evaluate the performance of existing ones. The input parameters required for the analysis are T_i, U, A, W, C_P, and T_s.

GENERALIZED EQUATIONS FOR COMPRESSORS

A process engineer's task is often to evaluate the performance of a compressor unit based on gas throughputs and terminal pressures. Since compressor stations are complex machines and operations, the analysis required is sophisticated and goes well beyond simple computations on a personal computer, although some preliminary evaluations can certainly be made. In this section we summarize the working expressions for standard compressor operations. Compressor operations can be categorized under three thermodynamic categories:

1. *Isothermal.* When compression takes place at constant temperature. This situation

is strictly hypothetical. A constant-temperature operation can be approached only when the compressor runs at an infinitely slow speed.

2. *Adiabatic.* When the net heat lost or gained by the unit to or from the surroundings is zero. Most plant installations approach this operation, and the adiabatic equations are widely used.

3. *Polytropic.* Sometimes the compression process has certain associated irreversibilities. The actual operation is therefore approaching adiabatic, but not quite. This "approximately adiabatic" operation is called polytropic.

In evaluating and/or designing compressors the main quantities that need to be calculated are the outlet (discharge) gas temperature, and the energy required to drive the motor or other prime mover. The latter is then corrected for the various efficiencies in the system. The differential equations for changes of state of any fluid in terms of the common independent variable are derived from the first two laws of thermodynamics:

$$dQ = TdS = C_v dT + T\,(\partial P/\partial V)_V dV$$

$$dQ = TdS = C_p dT - T(\partial V/\partial T)_P dP$$

$$W_{cycle} = -\int V dP,$$

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where Wcycle refers to Cycle Work, and the integration is over the limits of P_1 to P_2. In the case of isothermal compression, for an ideal gas, we may state the following:

$$W = \int PdV = RT \int dv/v = -RT \int dP/P, \text{ or}$$

$$W = -RT \ln [P_2/P_1]$$

The above expression is valid for either compression or expansion between the given pressure limits. For the case of compression, $P_2 > P_1$, and the work is negative (i.e., work is done on the system). For expansion, $P_2 < P_1$, and the work term is positive (i.e., work is performed by the system). If compression is isothermal and the gas is ideal, then the cycle work per mole is also given by the last equation stated above.For adiabatic compression, the adiabatic change is described by the following equations, where k is the ratio of specific heats and some typical values for common gases are : 1.67 for monatomic gases (e.g., He, A, etc.); 1.40 for diatomic gases (e.g., H_2, CO, N_2); and 1.30 for tri-, tetra- and penta-atomic gases (e.g., CO_2, CH_4, etc.):

$$P_1V_1^k = P_2V_2^k = P_3V_3^k = ,\ldots$$

$$(T_2/T_1) = (P_2/P_1)^{(k-1)/k}$$

$$W_{Theoretical} = \{RT_1/(k-1)\}[1 - (P_2/P_1)^{(k-1)/k}]$$

$$W_{Cycle} = (NK)\ (W_{Theoretical})$$

For polytropic compression, the change may be described by the following equations:

$$P_1V_1^n = P_2V_2^n = P_3V_3^n = ,\ldots$$

$$(T_2/T_1) = (P_2/P_1)^{(n-1)/n}$$

$$W_{Theoretical} = \{RT_1/(n-1)\}[1 - (P_2/P_1)^{(n-1)/n}]$$

$$W_{Cycle} = (Nn)\ (W_{Theoretical})$$

Since non-ideal gases do not obey the ideal gas law (i.e., PV = nRT), corrections for nonideality must be made using an equation of state such as the Van der Waals or Redlich-Kwong equations. This process involves complex analytical expressions. Another method for a nonideal gas situation is the use of the compressibility factor Z, where Z equals PV/nRT. Of the analytical methods available for calculation of Z, the most compact one is obtained from the Redlich-Kwong equation of state. The working equations are listed below:

$$h = b/V = (0.0867RT_c)/VP_c$$

$$Z = [1/(1 - h)] - (4.934/T_r^{1.5})[h/(1 + h)]$$

And form PV = NZRT, we note that T = PV/NZR, and hence, at any initial state 1, we may write the following:

$$T_1 = (P_1V_1)/(NZ_1R)$$

Combining the above expressions results in the following equation for the cycle work required in an adiabatic change:

$$W_{Cylce} = kV_1/\{N(k-1)Z_1\}[P_1 - (P_2/P_1)^{(k-1)/k}]$$

Similarly, the cycle work required in a polytropic process is given by the following expression:

$$W_{polytropic} = nV_1/\{N(n-1)Z_1\}[P_1 - (P_2/P_1)^{(n-1)/n}]$$

For an isothermal change, the expression for P from the Redlich-Kwong equation can be substituted into the general formula for work done:

$$W = \int PdV$$

$$P = (RT)/(V - b) - (a)/(T^{1/2}V(V + b))$$

Substituting and combining above expressions we obtain:

$$W = \int [\{RT/(V - b)\}dV - adV/(T^{1/2}V(V + b)]$$

The above integral can be evaluated either analytically or numerically by applying Simpson's rule. The following provides a summary of the major working equations for compressor analysis:

For the discharge temperature in an adiabatic process:

$$T_2 = T_1(P_2/P_1)^{(k-1)/k}$$

For the discharge temperature in a polytropic process:

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$$T_2 = T_1(P_2/P_1)^{(n-1)/n}$$

For theoretical cycle work performed in an adiabatic compression cycle (nonideal fluid):

$$W = kV_1/\{N(k-1)Z_1\}[P1 - (P_2/P_1)^{(k-1)/k}];$$

$$Z = [1/(1 - h)] - (4.934/T_r^{1.5})[h/(1 + h)];$$

$$h = 0.0867\ RT_c/VP_c$$

For the theoretical cycle work performed in a polytropic compression cycle (nonideal fluid):

$$W = nV_1/\{N(n-1)Z_1\}[P_1 - (P_2/P_1)^{(n-1)/n}]$$

For theoretical cycle work performed in an isothermal compression cycle:
(For ideal fluid case)

$$W = -RT\ \ln(P_2/P_1)$$

(For nonideal fluid case)

$$W = \int[\{RT/(V - b)\}dV - adV/(T^{1/2}V(V + b)]$$

BATCH DISTILLATION: APPLICATION OF THE RAYLEIGH EQUATION

In discontinuous simple open distillation (batch distillation, Rayleigh distillation), the distillation still is charged with a liquid mixture (feed). Heating to the boiling point partially vaporizes the liquid. The vapor is condensed and collected in the distillate receiver. Refer to Figure 8 for a simplified definition of the operation. Batch distillation is a dynamic process. The composition of distillate and liquid remaining in the still (residue) as well as temperature change with time. Figure 9 illustrates these changes qualitatively. At any time during the distillation, the total number of moles in the still is n_R with mole fraction x_R of the considered component. After distilling an incremental amount dn_R with mole fraction y, the number of moles in the still is $n_R - dn_R$. We may now write a component material balance as follows:

$$ydn_R = n_R x_R - (n_R - dn_R)(x_R - dx_R)$$

Neglecting $dn_R dx_R$ as the product of two small quantities, the following differential equation is derived which relates the amount of residue in the still to the composition:

$$dn_R/n_R = dx_R / (y - x_R)$$

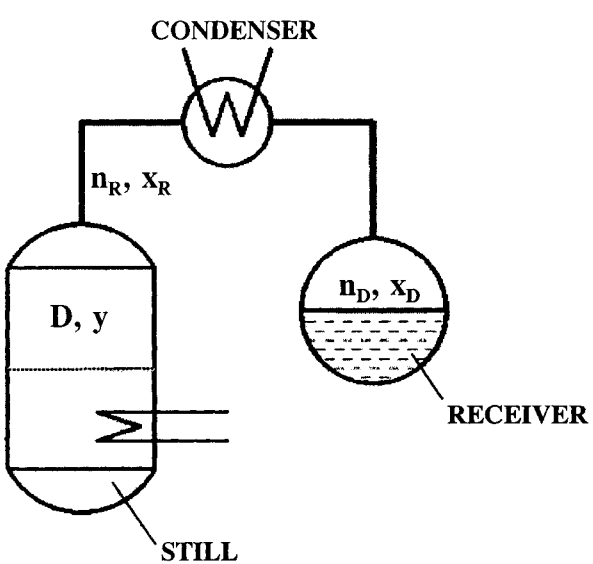

Figure 8. *Simple batch distillation.*

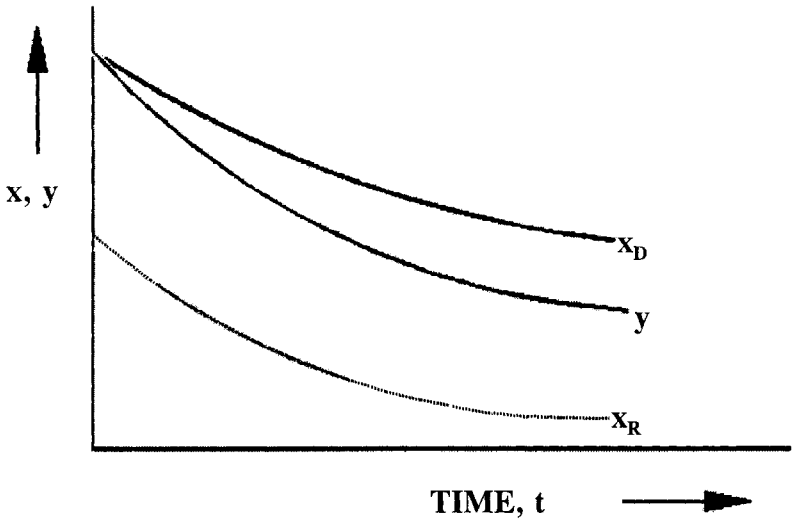

Figure 9. *Dynamic equilibrium curve for batch distillation.*

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This equation was first given by Lord Rayleigh and is called the Rayleigh equation. Integration between the initial number of moles n_{R0} in the still with composition x_R over any time yields the following:

$$\ln (n_R/n_{R0}) = \int 1/(y - x_R)\, dx_R$$

Note that this equation holds for any component in a multi-component mixture. The integral on the right-hand side can only be evaluated if the vapor mole fraction y is known as a function of the mole fraction x_R in the still. Assuming phase equilibrium between liquid and vapor in the still, the vapor mole fraction $y(x_R)$ is defined by the equilibrium curve. Agitation of the liquid in the still and low boilup rates tend to improve the validity of this assumption.

By using vapor-liquid equilibrium data the above integral can be evaluated numerically. A graphical method is also possible, where a plot of $1/(y - x_R)$ versus x_R is prepared and the area under the curve over the limits between the initial and final mole fraction is determined. However, for special cases the integration can be done analytically. If pressure is constant, the temperature change in the still is small, and the vapor-liquid equilibrium values (K-values, defined as $K=y/x$ for each component) are independent from composition, integration of the Rayleigh equation yields :

$$n_R/n_{R0} = [x_R/x_{R0}]^{1/(K-1)}$$

More often than the assumption of constant K-values, the assumption of constant relative volatilities is applied. The relative volatility of two components "i" and "j" is defined as the ratio of their K-values:

$$\alpha_{ij} = K_i / K_j = y_i x_i / x_i y_j$$

For a binary mixture with constant relative volatility the following expression applies:

$$\alpha = y(1 - x_R) / x_R(1 - y)$$

It then follows that:

$$\ln(n_R/n_{R0}) = \{1/(\alpha - 1)\}\{\ln(x_{R0}/x_R + \alpha\ln[(1 - x_R)/(1 - x_{R0})]\}$$

The total number of moles n_D and composition x_D in the distillate receiver can now be obtained from the material and component material balances:

$$n_D = n_{R0} - n_R$$

$$x_D = [n_{R0}x_{R0} - n_R x_R]/n_D$$

Assuming a specific boilup rate "D", the compositions may now be calculated as a function of time:

$$dx_R/(y - x_R) = -(D/n_R)dt$$

where

$$n_R = n_{R0} - \int Ddt$$

The above integration is performed over the limits of 0 to time t. Again, assuming D to be a constant, then:

$$\int dx_R/(y - x_R) = \ln[1 - Dt/n_{R0}]$$

Upon integration, we obtain the final working expression which can be readily programmed on to a spreadsheet for calculations:

$$x_R/x_{R0} = \ln[1 - Dt/n_{R0}]$$

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